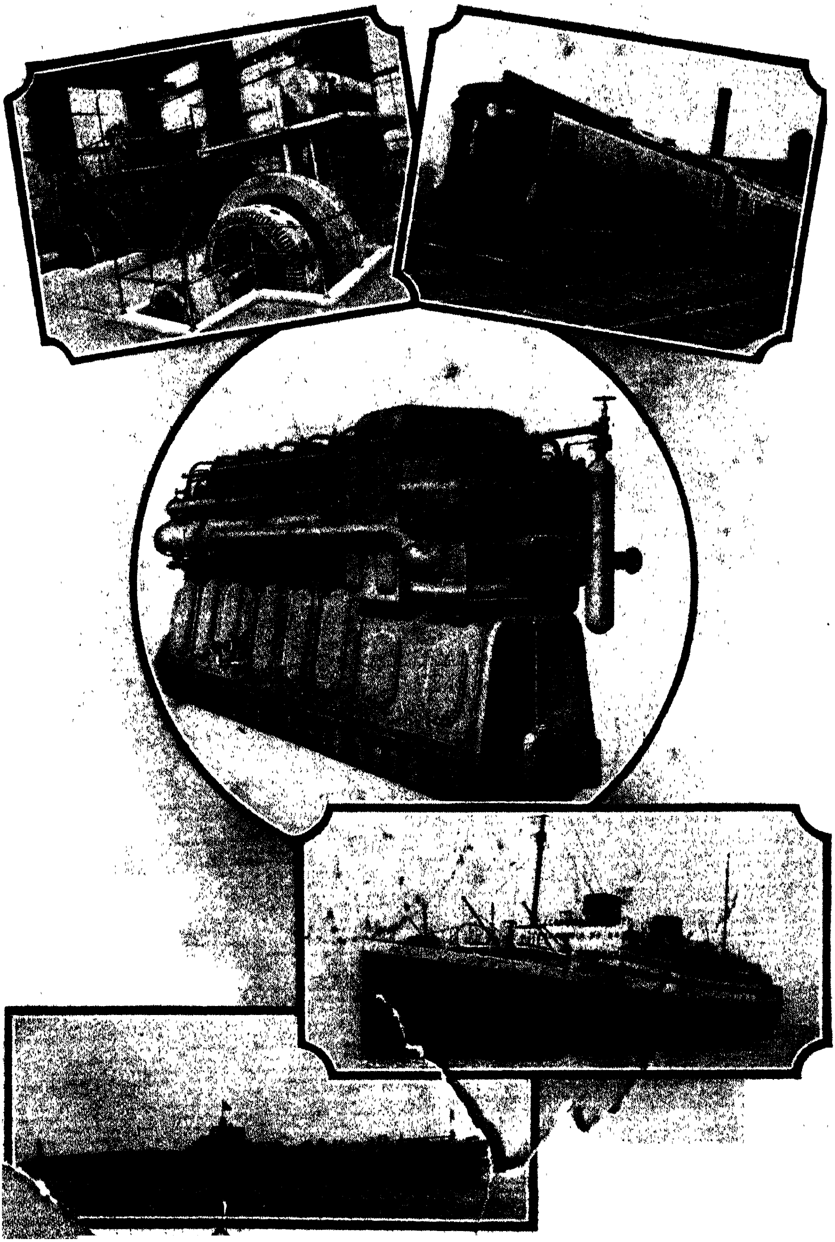

DIESEL ENGINES
Marine—Locomotive—Stationary

A PRACTICAL TREATISE

ON THE PRINCIPLE, CONSTRUCTION, OPERATION AND MAINTENANCE OF THE DIESEL OIL ENGINE, BOTH MARINE AND STATIONARY TYPES, WITH A DESCRIPTIVE CHAPTER ON THE LATEST DEVELOPMENTS IN DIESEL LOCOMOTIVES AND DIESEL ELECTRIC DRIVE FOR SHIP PROPULSION

BY

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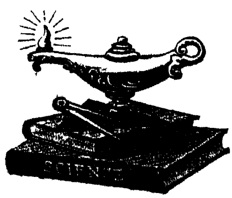

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TO THE

CHANWOS CLUB

PREFACE

THIS book was written and compiled with the object of presenting to the practical operating engineer the elementary principles, care and operation of the Diesel engine.

The men who do the actual operating of the Diesel engine, and are charged with its upkeep and repair, care, as a rule, very little about thermodynamics, theory and design of the internal combustion engine. Their desire for knowledge is more practical, such as will prove useful to them in their everyday work.

From a practical man's viewpoint, the great fault with most books on the Diesel engine is that in the second chapter they plunge deep into thermodynamics, and from that on to more theory and design. Such books are very useful and necessary to the mechanical engineer and machine designer, but the practical man is all but lost in their contents.

One of the greatest troubles of ship and power station owners today is in obtaining experienced Diesel engine operators. There are few men who thoroughly understand the Diesel engine, especially in America, where these engines have been more slow in developing. The man who thoroughly understands the workings, care and operation of the Diesel engine is coming to be in considerable demand. The old fashioned steam engineer is generally loath to change from the old to the new power.

In the U. S. Navy the lack of experienced Diesel engine operators is most keenly felt. Young inexperienced men are taken from the fleet and given a few months' instruction in submarines and Diesel engine operation and are then assigned to submarine duty. Owing to the constant coming and going of these young men and to their inexperience it is truly wonderful the results obtained. The engineering records of our submarines, taking into consideration the arduous duties aboard ship, their almost constant operation and the constant changing of personnel, is something which the manufacturers of these engines can well be proud of.

Preface

No engine is more susceptible to mishandling than the Diesel. There is a large number of somewhat intricate units of the machinery, each of which is vital to the working of the whole plant, and any one of which, if allowed to operate at less than a fair percentage of full efficiency, can cause considerable trouble. It is only fair to state that the necessary adjustments of these parts is as readily remedied as they are liable to require attention, provided, of course, that the engineers are as well trained in expert observance as they are versed in the means of making correct adjustments and repairs.

The Author hopes that this elementary practical work on the Diesel engine will prove useful to the practical operating engineer, and also prove interesting to the more advanced engineer and designer.

In concluding the Author wishes to thank the following manufacturers and publishers for their generous help and contributions:

Lubrication (Texas Oil Co.); *Oil Engine Power; Motor Ship; Railway and Locomotive Engineering;* Westinghouse Electric and Manufacturing Co.; General Electric Co.; New London Ship and Engine Co.; Bethlehem Shipbuilding Corp.; Worthington Pump and Machinery Corp.; Foos Gas Engine Co.; and *Power* for the 200 Diesel Engine Pointers.

Mr. H. E. Harrisson of McIntosh and Seymour Co. for his excellent contribution on the engines of their manufacture.

Lieutenant R. A. Demming, U.S.N., for his help and criticism in reading the manuscript and Lieutenant George Marvel, U.S.N., at whose suggestion this book was compiled.

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July, 1926

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INTRODUCTION

THE modern Diesel Engine is the outcome of various experiments with internal combustion engines in all parts of the globe.

Various scientists appear to have gone on record as to the use of gunpowder being fired in a closed vessel, as a source of energy, but Huyghens seems to be the first to build an engine of this nature in 1680. The design was simply an open ended cylinder with a piston and the powder was exploded when the piston was at the bottom of the cylinder; the explosion forced the piston upward. Structural defects in the design, however, caused its abandonment.

The various patents on record since are mostly of European inventors, with the exception of that recorded by George Bailey Brayton, a Philadelphian, in 1872.

This engine differed from all other internal combustion engines in that the charge was introduced into the cylinder so as to burn at constant pressure, as against the other designs wherein the charge burned at a constant volume. Its abandonment was chiefly because of its excessive size and weight, both from a manufacturing and operating standpoint, as compared to the engines of the "Otto" cycle, which were then fast coming to the fore at that time.

It was twenty years later, in 1892, that Dr. Diesel set forth a theory of an engine which should be so designed as to insure the greatest economy possible, both as to use and the consumption of fuel. The object he proposed to attain was by the following distinctive features of operation:—The attainment of the highest temperature of the cycle before combustion took place; this high temperature being generated solely by the compression of air; the introduction of fuel into this high temperature by slow degrees at the commencement of the expansion stroke in such a manner that combustion at constant temperature could be maintained, and lastly, by such an adjustment of the properties of air and fuel that no cooling water would be required.

Dr. Diesel's ideas were patented some time before his theory was published and two very important German engineering firms, *Krupp* and *Maschinenfabrik-Augsburg-Nurnberg*, took up the development of his patents. After many difficulties, the first mentioned firm produced a successful engine and so far as the combustion of the fuel was concerned, it proved to be the most economical internal combustion engine that had been produced to that date. This resulted in a number of other firms taking up the manufacture of the Diesel engine under royalty, and it is to their efforts, their failures and successes, both in Europe and the United States, that we owe the comparatively perfect engine of today.

The first engine of this type built in the United States was completed in 1898 at the plant of Adolphus Busch in St. Louis, Mo. He purchased outright the sole rights to all Dr. Diesel's existing and future patents in the United States, and so introduced the Diesel engine into this country. Its superiority was soon recognized owing to its high efficiency and its adaptability to general power requirements. The demand for engines for stationary power plants of larger and larger sizes developed rapidly, and its possibilities for ship propulsion were also quickly realized and developed, and a few years found the wasteful and more expensive steam installations being supplanted by it, both in Marine and Stationary power fields.

Some of the savings effected by these engines when changing a station over from steam to Diesel are surprising as well as interesting. The general idea of the efficiency of a steam plant is usually gained from what is published from time to time in the technical press with regard to the various large and efficient stations that have been erected throughout the country, the fact that most of the power generated is generated in stations of quite moderate size being overlooked.

Generally speaking the size of a station depends upon the needs of the community which it is intended to serve and without a doubt these small stations are very different from the large and highly efficient generating stations usually mentioned and lauded in the technical press. The one or two instances given below will serve to drive home this fact.

The City of Woodward, Oklahoma, installed during the

year 1922 two 500 H.P. Diesel engines and careful records having been kept of the steam plant, which the Diesels displaced, it is easy to get the necessary data for this comparison. On an audit covering a period of 395 days immediately prior to the closing down of the old steam plant, it was found that the coal used every day cost $119.05 delivered at the plant. Fuel oil was also used which cost $8.56 per day. Packing and waste, etc., amounted to $1.58 per day, while their expenses actually paid for repairs including shop work, welding flues and grates for the boilers, but not including any new piece of machinery, was over $40.00 per day. During this period 19 men were employed at the plant all of the time and 21 men part of the time.

After the installation of the Diesel plant it was found that the two engines installed were producing more current than the old steam engines, the cost of fuel per day was less than $15.00, lubricating oil was less than $1.50 per day and 7 men only were employed, including a chief engineer, 3 shift engineers and 3 helpers.

Taking a more direct comparison as during the month of December, 1922, when the Diesel plant was running, it was shown that the fuel cost for the power station for the month was $401.50, lubricating oil was $27.30, making a total of $428.80, while the cost of fuel and lubricating oil only for the previous December, that is 1921, when the steam plant was running, was $3,285.00.

Another instance was that of the installation of a 750 H.P. Diesel engine where fuel oil had been used under the boilers instead of coal and Messrs. Stone & Webster, Inc., reported that the Diesel engine generated between 75 and 80% of the total station output in kilowatt hours, and that in comparison with the operation of the previous year, there was obtained ten times the kilowatt hours output from the same quantity of the same kind of fuel oil by using the Diesel engine, as was generated by the steam plant. The comparative results just quoted certainly make a good showing in favor of the Diesel engine and while it is not possible to give definite comparisons, it is worth while recording the operating costs obtained at the city of Colby, Kansas.

At the city of Colby, where two 200 H.P. units are installed, the operating costs for the five months from July 1, 1922 to November 1, 1922, was for fuel $1,011.15, for lubricating oil $144.62, repairs $2.52, the K.W. hours produced 221,750. The load factor for the station works out from the above at approximately 27% with a cost of fuel per K.W. hour of 46c.

In the marine world the savings are equally good. The ships "Covedale" and "Courtois" when operating as steam vessels under the Shipping Board, consumed 28 tons of oil per day under their boilers in each case. They were converted to motor ships and re-named "Muncove" and "Munmotor" respectively, the "Muncove" being fitted with Diesel-Electric auxiliaries as well as Diesel main drive, while the "Munmotor" kept her steam auxiliaries.

The fuel consumption of the "Muncove" was reduced to 4½ tons per day, while that of the "Munmotor" was reduced to 8½ tons per day. Later the "Munmotor's" auxiliaries were changed to operate upon compressed air, the air being taken from the surplus air of the main compressor, and the consumption was then brought down to 6¼ tons per day.

As regards reliability and their ability for long and continuous service, non-stop runs of 201 and 190 days, mostly under full load, have been registered, without any apparent harmful results, while it has become quite common practice, in many places, to run for 90 days without a shutdown of any kind, and in the marine service there is the Motor ship "Bayonne" up to August, 1925 logging 351 round trips of 428 nautical miles each, without any delays or detention whatever, other than the annual inspection for insurance purposes, while the owners of the Motor ship "Carolyn Francis" boasted in 1925 of having had the vessel in the arduous service of the Arctic circles since 1920, and never having had to spend so much as $1.00 for repairs.

With such results as these being constantly obtained from the various makes of engines in service, it is not to be wondered at, that the old fashioned steam engine is rapidly being displaced in favor of the more modern Diesel Engine Units.

DIESEL ENGINES

CHAPTER I

Elementary Thermodynamics

The Unit of Heat—The Unit of Work—The Unit of Power—Thermodynamic Relations—Specific Heat—Expansion of Gases—The Diesel Cycle—Theoretical Diesel Indicator Card—Mechanical Efficiency—Fuel Economy—Thermal Efficiency—Suction Heating—Definition of Cycle—Definition of Diesel Engine.

In taking up the study of the Diesel engine it is important to become familiar with a certain amount of theory regarding the behavior of the working substances in the cylinder. While it is, of course, essential from a designer's point of view to make an exhaustive study of the thermodynamic relations, it is believed that, for the operating engineer, a brief summary of the outstanding principles will suffice. For a more elaborate study of the theory underlying Diesel engine design, the reader is referred to any standard textbook on thermodynamics.

From the thermodynamic point of view, the fundamental properties of a gas are its pressure, volume and temperature. With a knowledge of these properties, all the other existing relations may be found. Before discussing these three properties it is necessary to define the units employed.

The Unit of Heat. The unit of heat is the British thermal unit (B.T.U.), and is that amount of heat required to raise the temperature of one pound of water one degree from a temperature of 62 to 63 degrees Fahrenheit.

The Unit of Work. The unit of work is the foot-pound, and is the work done in lifting a weight of one pound through a vertical distance of one foot.

The Unit of Power. The unit of power is the horse power and is equal to 33,000 foot-pounds per minute. Thus, to convert work into power, divide the work (foot-pounds) by 33,000, the result being horse power.

Heat units are convertible into work units by means of the mechanical equivalent of heat, which means that one B.T.U. is equivalent to 778 foot-pounds. This quantity is usually called the Joule, in honor of its discoverer.

In all thermodynamic relations it is customary to express the temperature on the absolute scale. Thus, in Fahrenheit units, the absolute temperature equals the degrees F. plus 460. Likewise, all pressures are expressed in pounds per square foot and all volumes in cubic feet.

Specific Heat. The specific heat of a substance is the amount of heat in B.T.U.'s which must be supplied to one pound of the substance to raise its temperature from 62 to 63 degrees F.

For all gases there are two specific heats,—

(1) At constant volume, C_v.

(2) At constant pressure, C_p.

If one pound of a gas be heated in a closed container until its temperature is raised one degree F., a certain amount of heat is necessary. There is no volume change and consequently no external work is done. The heat added increases the internal energy of the gas, raises its pressure, and its temperature one degree. The quantity of heat required to do this is called the specific heat of the gas at constant volume, Cv.

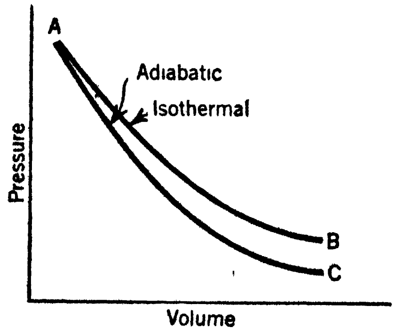


Fig. 1.—Expansion

If this same pound of gas were heated in a cylinder fitted with a piston by means of which the pressure of the gas could be kept constant, it would be found that a greater amount of

heat is required to raise the temperature of the gas one degree F., and as the volume of the gas increased during the heating the piston is forced out to keep the pressure constant. In this case the heat added, besides increasing the internal energy of the gas and raising its temperature one degree, also performs external work in forcing the piston out against constant pressure. The quantity of heat required under these conditions is called the specific heat of the gas at constant pressure, Cp. The external work done in moving the piston accounts for the difference between C_p and C_v.

For pure air C_p equals 0.2389
C_v equals 0.1704

For adiabatic expansion and compression of air:

$$\frac{C_p}{C_v}=S=\frac{0.2389}{0.1704}=1.41$$

Expansion of Gases. A gas may be expanded or compressed in one of two different ways; Isothermally and Adiabatically.

Isothermal. In which the gas is compressed or expanded at constant temperature.

Adiabatic. In which the gas is compressed or expanded so that it does not gain or lose heat.

In a Diesel engine, the expansion and compression of the gas occurs in such a short period that it has very little opportunity to gain or lose heat, and consequently the expansion is practically adiabatic in all cases.

Fig. 1 shows a gas of original condition of pressure and volume at the point A which has been expanded isothermally along AB, and adiabatically along A-C. Fig 2 shows a gas of original pressure and volume at A which has been compressed isothermally along A-B, and adiabatically along A-C. It will be seen that adiabatic expansion and compression gives a more rapid rate of pressure change than isothermal.

The Diesel Cycle. It is desirable to have some ideal standard for the comparison of the actual working of one engine with that of another. For this purpose the Constant Pressure or Diesel cycle is used. Fig. 3 shows the Diesel cycle which consists of adiabatic compression along 1-2, combustion of

fuel along 2-3 at constant pressure, adiabatic expansion along 3-4, and exhaust to constant volume along 4-1.

This theoretical cycle card, or indicator diagram, is worked

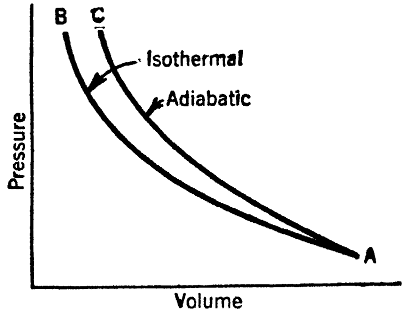


Fig. 2.—Compression

by air throughout, and therefore differs considerably from the actual indicator cards taken on a Diesel engine.

The pressure, volume and temperature at any point on the theoretical card can be found by calculation by the following formulae:

Along the horizontal line 2–3,—

Pressure is constant. $\frac{V_2}{V_3} = \frac{T_2}{T_3}$

Along the vertical line 4–1

Volume is constant. $\frac{P_4}{P_1} = \frac{T_4}{T_1}$

Along the adiabatic lines, 1–2 and 3–4,—

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{S-1}{S}} = \left(\frac{V_2}{V_1}\right)^{S-1}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{S-1}{S}} = \left(\frac{V_4}{V_3}\right)^{S-1}$$

Fig. 4 shows the theoretical Diesel diagram where the expansion has been extended to atmosphere. In actual practice, the "toe" of the diagram is cut off, as the extended expansion

as shown in the drawing would require a very large cylinder. Cutting off the "toe" gives greater mean pressure and more power for an equal engine weight. The small gain in the total work, as represented by the shaded area, is more than compensated for by the saving in the size and weight of the cylinder.

Mechanical Efficiency. The mechanical efficiency (Em) of an engine is the ratio of the work delivered by the shaft to that supplied at the pistons in the form of heat,—

$$E_m = \frac{\text{B.H.P.}}{\text{I.H.P.}}$$

$$E_m = \frac{(\text{I.H.P.}) - (\text{Friction H.P.})}{\text{I.H.P.}}$$

where friction horse power is the horse power required to cover the losses due to mechanical friction of the moving parts, fluid friction of the working gases, the work of charging and exhausting, power to drive the spray air compressor and fuel pump, and, with the 2-cycle engine, the pre-compression or pump work of scavenging. The average mechanical efficiency of a correctly designed and operated Diesel engine is from 73

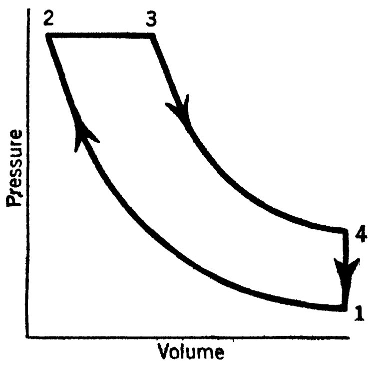


Fig. 3.—The Diesel Cycle

to 80 percent for the 4-cycle type, and from 60 to 70 percent for the 2-cycle engine.

The losses occurring between the supply of heat to the engine in the form of fuel and the delivery of heat in the

form of work at the shaft are divided into three parts, and the ratios of the heat used to the heat supplied at each step are,—

(1) Efficiency of the cycle=Et
(2) Card factor=Ed
(3) Mechanical efficiency=Em

Fuel Economy. The fuel economy or fuel consumption of a Diesel engine is the consumption of fuel in pounds per brake horse power hour (B.H.P.H.). The average fuel consumption of a well designed engine using oil containing 18,500 B.T.U.'s per pound is about .42 pound per B.H.P.H.

Thermal Efficiency. The thermal efficiency of the Diesel cycle is the ratio between the amount of heat transformed into external work divided by the total heat supplied to the cycle. The thermal efficiency or cyclic efficiency of the ideal Diesel cycle is given by the equation,—

$$E_t = \frac{\text{(heat added)} - \text{(heat rejected)}}{\text{heat added.}}$$

Referring to Fig. 3—

$$E_t = 1 - \frac{T_4 - T_1}{S(T_3 - T_2)}$$

using temperatures.

$$E_t = 1 - \left(\frac{1}{(r)^{S-1}} \times \frac{Y^{S-1}}{S(Y-1)}\right) = \text{Theoretical thermal efficiency}$$

where.—

$$r = \frac{V_1}{V_2} = \text{Volumetric compression ratio.}$$

$$Y = \frac{V_3}{V_2} = \text{Ratio of cut-off volume to clearance volume.}$$

$$S = \frac{C_p}{C_v} = 1.41$$

This equation shows that the Diesel cycle efficiency varies (1) directly with a certain power of the compression ratio,

(2) directly with the value of S, and (3) inversely as some function of the injection period, or cut-off.

The thermal efficiency of the Diesel cycle is the same as that of the Otto cycle except for the factor,—

$$\frac{Y^{S-1}}{S(Y-1)}$$

The efficiency of the Diesel cycle therefore depends not only on the compression, but also upon Y, that is, volume at the end of combustion. The earlier in the stroke the fuel is cut off, the smaller will be the value of Y, and the greater the thermal efficiency.

The cut-off ratio exercises an important influence upon the thermal efficiency of the Diesel cycle. This is borne out in practice since Diesel engines show high efficiency for high compression, and also operate efficiently at partial loads or at

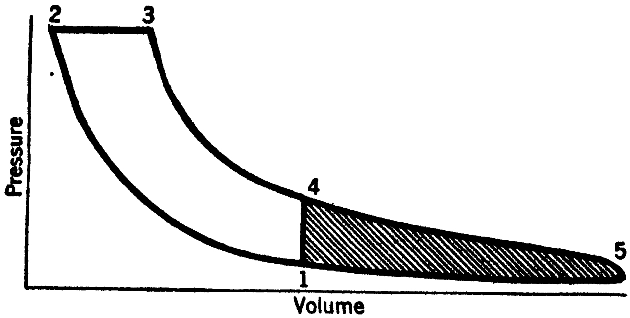


Fig. 4.—Complete Expansion of Gases

reduced power. A large number of tests of Diesel engines have shown a greater thermal efficiency at three-quarters load than at full load. This does not hold good for still lighter loads because of other factors.

The indicated thermal efficiency, (Ei) is as follows,—

$$E_i = \frac{2545 \times (\text{I.H.P.})}{\left\{\begin{matrix}\text{Total fuel consump-}\\ \text{tion per hour.}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Calorific value of fuel in}\\ \text{B.T.U.'s per pound}\end{matrix}\right\}}$$

where the I.H.P.=2545 B.T.U. per hour.

Card Factor. Let a theoretical card be constructed from an actual engine, that is, a cycle having the same pressure and volume limits as those on which the engine works. If an actual indicator card is taken with the engine in operation and superposed on the theoretical card described above, there will be a discrepancy as shown in Fig. 5.

A study of this figure shows that the engine is actually producing only about one-half of the work which the theoretical card shows possible. The ratio of the two areas (actual card area divided by the theoretical card area) is called the Card Factor, or Diagram Factor.

$$\text{Card Factor} = \frac{\text{Indicated thermal efficiency}}{\text{Theoretical thermal efficiency.}}$$

The losses which cause this reduction in the output of the engine are:

1. Imperfect ignition.
2. After-burning.
3. Radiation.
4. Heat carried off by cooling water.

The first three losses can not be separated, but the heat lost to the cooling water may be found by the equation:

B.T.U.'s lost to cooling water $= W \ (T2 - T1)$.

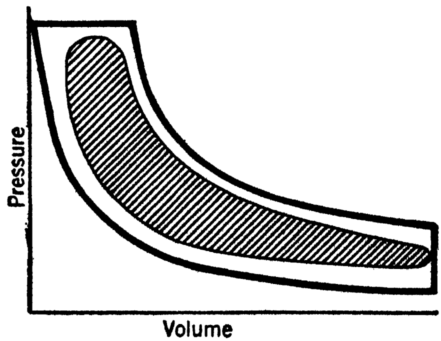


Fig. 5.—Comparison of Theoretical and Practical Cards

Where W=weight of cooling water used in pounds per hour.

T1=temperature of cooling water entering engine.

T2=temperature of cooling water leaving engine.

Volumetric Efficiency. The volumetric efficiency is the ratio between the weight of air in the cylinder of a 4-cycle engine at the commencement of the compression stroke and that required to fill the same volume with air at atmospheric pressure. If an engine has large valves and operates at a low speed it will be better able to suck in a full charge of air during the suction stroke and therefore will have a larger volumetric efficiency than an engine having smaller valves or running at a higher speed.

During the exhaust stroke a pressure a little above atmosphere exists in the cylinder, being necessary in order to drive out the exhaust gases through the valves, pipes and muffler. For this reason the exhaust line 3-4, Fig. 6 is above the atmosphere line. At the end of the exhaust stroke, the gases remaining in the clearance volume expand adiabatically along the line 4-5-6, cutting the atmosphere line at 5, until the pressure in the cylinder is lowered sufficiently for the outside air pressure to force a charge of air into the cylinder against the air friction resistance of the inlet valves and pipes. At the end of the suction stroke 7, the cylinder is full of air but at a pressure below atmosphere, and as the piston moves up, the charge is compressed along the line 7-8 before atmospheric pressure is reached. The distance 5-8 along the atmosphere line represents the volume of air at atmospheric pressure drawn into the cylinder. The ratio of this volume to the cylinder displacement is the apparent volumetric efficiency (Ev).

$$E_v = \frac{V_8 - V_5}{V_7 - V_4} = \frac{V}{D}$$

During the suction stroke there is always pre-heating of the air due to its contact with the hot valves, pipes, cylinder walls, and hot gases remaining in the cylinder clearance space. This suction heating reduces the charge weight of the air, and makes it less than that represented by the distance V8-V5 at atmospheric pressure and temperature. This decrease in the weight of the air charge decreases the amount of fuel that can be efficiently burned in the cylinder, and therefore reduces the power of the engine. The effect of suction heating of the

air charge weight is indirectly proportional to the absolute temperatures of the air before and after heating. Suction heating ranges between 100 and 300 degrees, with an average value of about 200 degrees.

$$\begin{pmatrix}\text{Atmospheric}\\ \text{temperature}\end{pmatrix} + \begin{pmatrix}\text{Suction}\\ \text{heating}\end{pmatrix} = \text{Final air temperature.}$$

The true volumetric efficiency equals the apparent volumetric efficiency multiplied by atmospheric temperature, absolute, and divided by the absolute temperature at the beginning of the compression stroke. Assuming that

$$\text{(Apparent) } E_v = \frac{V}{D} = .80$$

and that atmospheric temperature equals 60 degrees F., or 520 degrees absolute, and that suction heating equals 200 degrees, then final air temperature equals 520 plus 200 equals 720 degrees F., absolute. Then

$$\text{(True) } E_v = .80 \times \frac{520}{720} = .578$$

If, in this example, the cylinder when completely filled with air at atmospheric temperature and pressure is able to burn an amount of fuel equal to 100 B.T.U., with a thermal efficiency of 27 percent, then 27 B.T.U. are turned into useful work every cycle. However, due to the reduced weight of the air charge, as shown by the true volumetric efficiency, only .578 times 100 or 57.8 B.T.U. can be burned per cycle, and the useful work obtained equals .27 times 57.8 or 15.6 B.T.U. Therefore, for the same size and weight of engine, the power developed is little more than one-half, and the weight of the engine per horse power is almost doubled.

In the 2-cycle Diesel engine the same situation exists, only it is more aggravated. The scavenging efficiency of a 2-cycle engine is the ratio between the weight of air in the cylinder at the beginning of the compression stroke and the weight of the mixture of air and burnt gases.

Definition of "Cycle". A cycle is a series of events which occurs in the same order, over and over again.

Quite a number of internal combustion engines have been invented in the past, using a different series of actions or events within the cylinder between the power strokes, to get the fuel and air charge into the cylinder, and to burn or explode them. Most of these engines were named after their inventors and are said to operate upon the "Lenoir cycle," the "Clerk cycle", the "Diesel cycle", etc.

They all operate upon either one of two systems. In one system the various events of introducing the charge of air and gas, burning or exploding the same, and getting rid of the burnt gases is made to occur within two revolutions, or with four strokes of the piston. The other system accomplishes

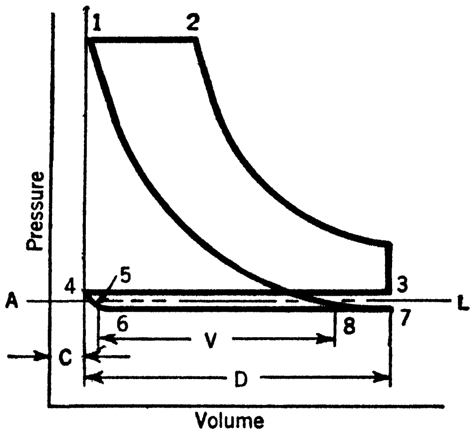


Fig. 6.—Volumetric Efficiency

C = Clearance Volume of Cylinder
D = Displacement Volume of Cylinder
V = Volume of Air at Atmospheric Pressure Actually in Cylinder
D — C = Total Volume of Cylinder

its series of events in one revolution, or with two strokes of the piston. The former system is called the four-stroke cycle and the latter the two-stroke cycle.

It should be understood that there are not four cycles in a four-stroke cycle engine and that each stroke of the piston is one cycle. An engine operates upon only one cycle which is accomplished in either two or four strokes of the

piston. When a Diesel engine is said to operate on the four-cycle principle, it means that it operates on "The Diesel principle in a cycle of four strokes." The terms "four-stroke cycle" and "two-stroke cycle" are commonly spoken of as "four-cycle" and "two-cycle, or "four-stroke" and "two-stroke."

Definition of a Diesel Engine. A Diesel engine is an internal combustion engine actuated by the gases resulting from the combustion of a liquid fuel injected in a fine state of sub-division into the engine cylinder at or about the conclusion of the compression stroke. The heat resulting from the compression of air to a high temperature is the sole means of igniting the charge of fuel. The combustion proceeds at, or approximately at, constant pressure.

Diesel engines are divided broadly into two types or classes, the four-stroke cycle and the two-stroke cycle. With both types the cylinder is filled with air at atmospheric pressure and the air is compressed by the piston to about 450 to 500 pounds pressure per square inch. This rapid compression of the air raises its temperature to about 1,100 degrees F., which ignites the fuel upon its being injected into the cylinder slightly before the piston reaches its top stroke. The oil burns rapidly, but without explosion, and the pressure exerted by its expansion produces the power impulse on the piston. The fuel is broken up into a fine spray and is injected into the cylinder during about one-tenth of the piston stroke, producing a fairly constant pressure nearly equal to the compression pressure at the end of the compression stroke.

CHAPTER II

Elementary Principles

The Diesel Four-Stroke Cycle—Function of Spray Valve—Function of Fuel Measuring Pump—Spray Air Compressor—Valve Actions and Valve Timing—The Diesel Two-Stroke Cycle—Scavenging—Port Scavenging Type—Overhead Scavenging Types—Valve Controlled Port Scavenging Types—Opposed Piston Engines—Double-Acting Engines—Surface Ignition Engines.

The Diesel Four Stroke Cycle. The 4-cycle Diesel oil engine with its various essential connections and auxiliaries is shown in diagrammatic form in Fig. 7.

The engine consists of one or more working cylinders, in which a piston moves up and down and is connected to the crankshaft by a connecting rod as shown. In the head of each working cylinder are three important valves; the spray valve, the inlet, and the exhaust valve. There are also other valves, such as the air starting and relief valves, which will be discussed later.

The function of the spray valve is to introduce a small quantity of fuel into the cylinder in the form of a fine spray, and this is accomplished in a very small fraction of a second. The fuel-measuring pump supplies the spray valve once each working cycle with a small charge of fuel which is delivered while the spray valve is closed. At the proper instant the spray-valve needle is lifted by suitable valve gear and the charge of fuel is blown into the cylinder by means of a powerful blast of high pressure air.

The compressed air for injecting the fuel into the cylinder is supplied by the spray air compressor which is driven direct from the main crankshaft. The compressor is of the two-stage type. Air is compressed in the first stage cylinder to about 90 pounds pressure per square inch. This compression raises the temperature of the air to about 350 degrees F., and after passing through the first stage cooler, it is delivered to the second stage cylinder at slightly above atmospheric tem-

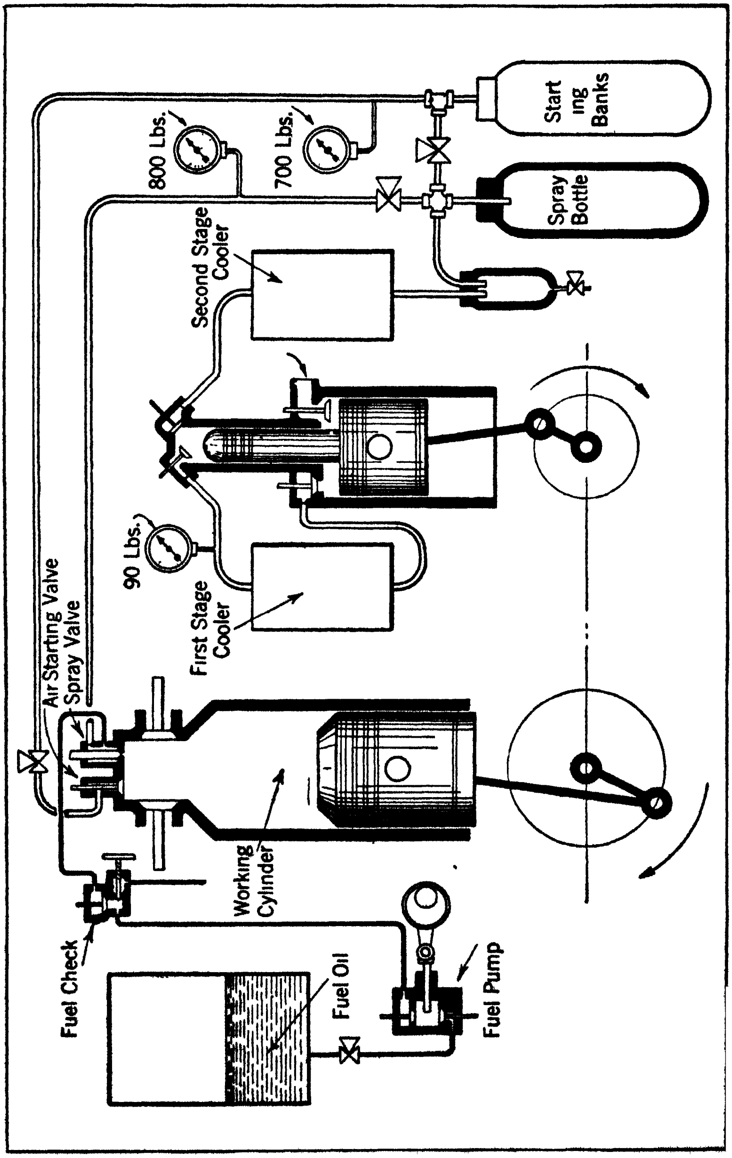


Fig. 7.—Diagrammatic Arrangement of 4-Cycle Diesel Engine

perature. The air is again compressed in the second stage cylinder to from 600 to 1,100 pounds pressure per square inch. After passing through the second stage cooler and the separator, which separates the moisture and oil from the air, it is delivered to the spray valves through a line called the spray air header.

Referring to Fig. 8 the action of the engine (the valve timings are from a Nelseco engine) will be seen as follows:

(A) On its downward, or admission stroke, the piston draws a charge of fresh air into the cylinder. The inertia of the air, due to its high velocity through the inlet valve, permits the valve to remain open until 20 degrees after bottom center, and causes a slight overpressure at the commencement of compression. In other words, at the commencement of compression, the cylinder contains a greater weight of air than it would contain at atmospheric pressure.

(B) At 20 degrees past bottom center, or 200 degrees around the crank path, the inlet valve closes and the piston goes up on its compression stroke, compressing the air to about 450 pounds pressure per square inch, which results in the air attaining a temperature of about 1100 degrees F. This is sufficient to ignite any liquid fuel which may be injected into the cylinder.

(C) At 7 degrees before top center on the compression stroke, the spray valve opens, and the spray air pressure, which is several hundred pounds greater than the compression in the cylinder, will start to blow into the cylinder the fuel contained in the spray valve. The fuel is injected in the form of a fine spray, or fog, and as it strikes the hot air in the cylinder it will immediately start to burn, and will burn continuously as long as the fuel is being sprayed in. The spray valve is held open during a period of 49 degrees, or until 42 degrees past top center. The temperature during this part of the stroke rises to about 2700 degrees F., and the pressure remains almost constant.

(D) After the spray valve closes and the fuel injection stops, the pressure in the cylinder will continue to drive the piston down, the pressure, of course, dropping as it expands and at the same time cooling off. When the piston is well

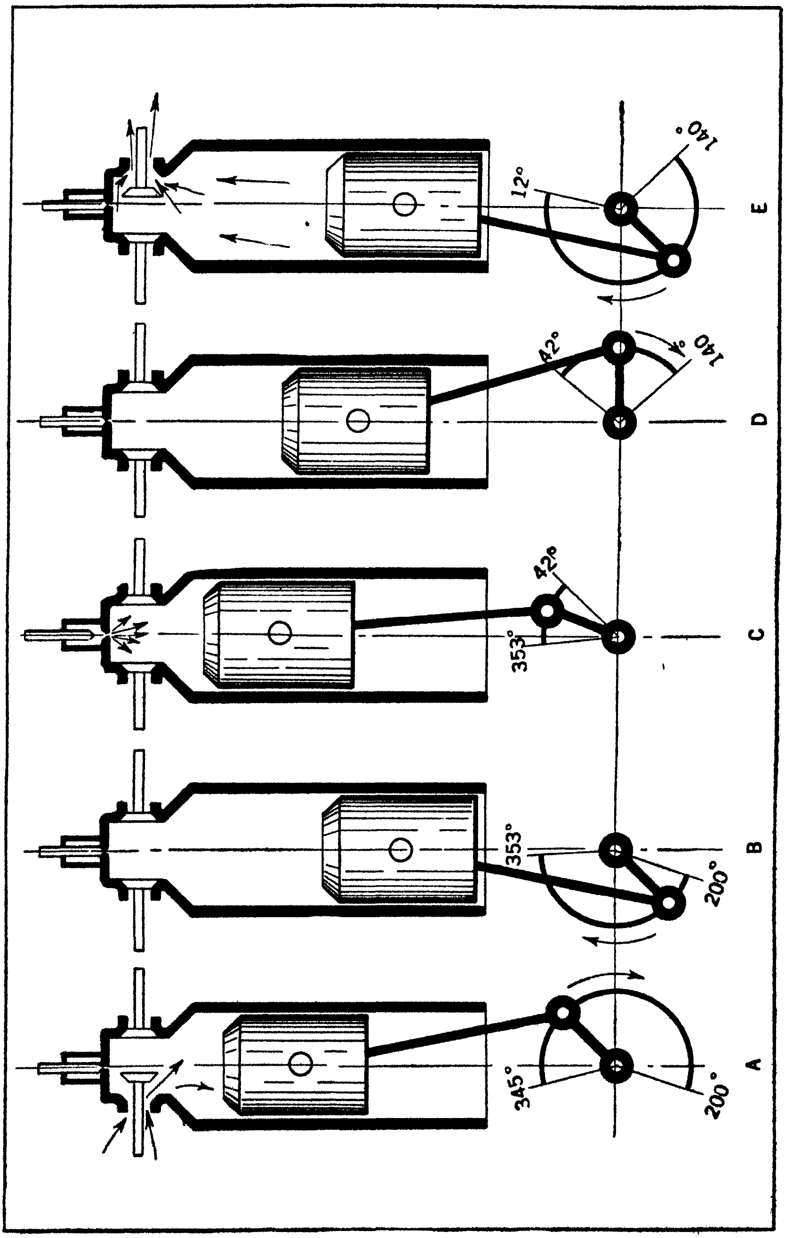


Fig. 8.—Valve Events of 4-Cycle Diesel Engine

down on its expansion stroke and the pressure has dropped to about 40 pounds and the temperature to about 700 degrees, the exhaust valve is opened and the exhaust gases start to escape.

(E) The exhaust valve opens at 140 degrees past top center, and before the piston reaches bottom center the pressure is relieved. On the up-stroke the burnt gases and products of combustion are forced out through the exhaust valve which remains open until 12 degrees past top center. Between 345 degrees and 12 degrees, both the inlet and exhaust valves are open. The inertia of the exhaust gases in passing through the exhaust valve and passages at a high velocity, makes it possible to keep the exhaust valve open until the piston has passed the upper dead center, which allows all the exhaust gases to escape.

It will thus be seen that there is a power impulse in each cylinder of a 4-cycle Diesel engine every other revolution of the crankshaft, or, with every fourth piston stroke. These four strokes are called the suction, compression, combustion, and exhaust strokes, and follow each other in the order named.

The power to drive the piston up and down on the other strokes beside the combustion, or power stroke, is obtained by two means, by the momentum of the fly wheel of the engine, and with an engine having several cylinders, the cylinders are so arranged that one of them is firing and furnishing power to the common crankshaft while the others are on one of their powerless strokes.

The engine is first started by compressed air at upwards of 350 pounds pressure per square inch. With four-cylinder engines, two cylinders are usually arranged to be operated by compressed air, which acts on the pistons in much the same manner as steam in a steam engine. The air passes at the proper time through the air starting valves, which are operated by means of suitable valve gear, and the engine turns over until the other two cylinders start to fire. The air starting mechanism is then cut out and all cylinders proceed to operate on fuel. Large marine engines, and those not having reversing clutches, usually have all cylinders fitted with air starting and reversing valves so that the engine can be started

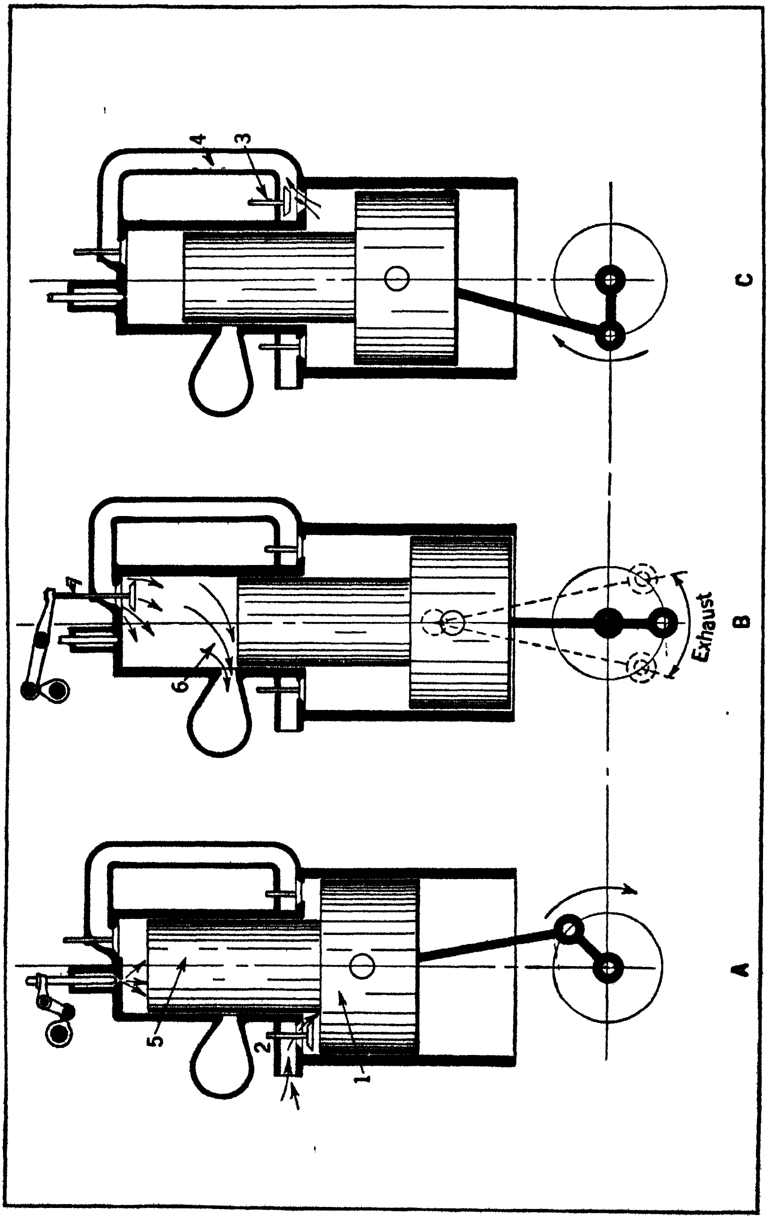


Fig. 9.—Diagrammatic Arrangement of 2-Cycle Step Piston Type Diesel Engine

and manoeuvered from any position without the dangers of its being caught on a dead center.

The Diesel Two-Stroke Cycle. As the name implies, the two-stroke cycle is accomplished in two strokes of the piston or in one revolution of the crankshaft. The cyclic events may be roughly divided into three nearly equal parts:

1. Combustion and expansion.
2. Exhaust and scavenge.
3. Compression.

Fig. 9 shows in diagrammatic form the elements of a 2-cycle Diesel engine. This engine is known as the Nürnberg or step-piston type. Each of the working cylinders has its own scavenging piston and cylinder which supplies air at about 10 pounds pressure to scavenge the working cylinder of its burnt gases and to charge the cylinder with fresh air. The action is as follows:

The scavenging piston:

(A) As the scavenging piston, 1, moves downward, pure air is drawn into the scavenging cylinder through the suction valve, 2, which is in communication with the atmosphere.

(C) As the scavenging piston moves upward this air is compressed in the scavenger cylinder to a pressure of about 10 pounds per square inch; the scavenging cylinder discharge valve, 3, which is held on its seat by spring pressure, automatically opens and the air discharges into the scavenger receiver, 4.

The working piston:

(A) The air which was previously charged into the working cylinder is compressed on the upstroke of the working piston to about 450 to 475 pounds pressure, and attains a temperature of about 1200 degrees F. Just before top center the spray valve opens and the fuel is injected into the cylinder, where it ignites and expands, forcing the piston down on its working stroke.

(B) As the piston nears its bottom stroke it uncovers an exhaust port, 6. Shortly after, the scavenger valve, 7, in the head of the working cylinder is opened by its cam and rocker arm, and the scavenging air from the scavenger receiver passes through into the cylinder and blows the burnt products

of combustion out through the exhaust port, 6, and fills the working cylinder with a charge of fresh air.

(C) As the working piston moves upward, it covers the exhaust port, after which the scavenger valve closes. Continuing upwards the piston again compresses the air and the cycle is repeated.

Port-Scavenging Type. Fig. 10 shows a diagrammatic arrangement of a 2-cycle, port-scavenging engine of the simplest type. In this system, the scavenging air, which is furnished

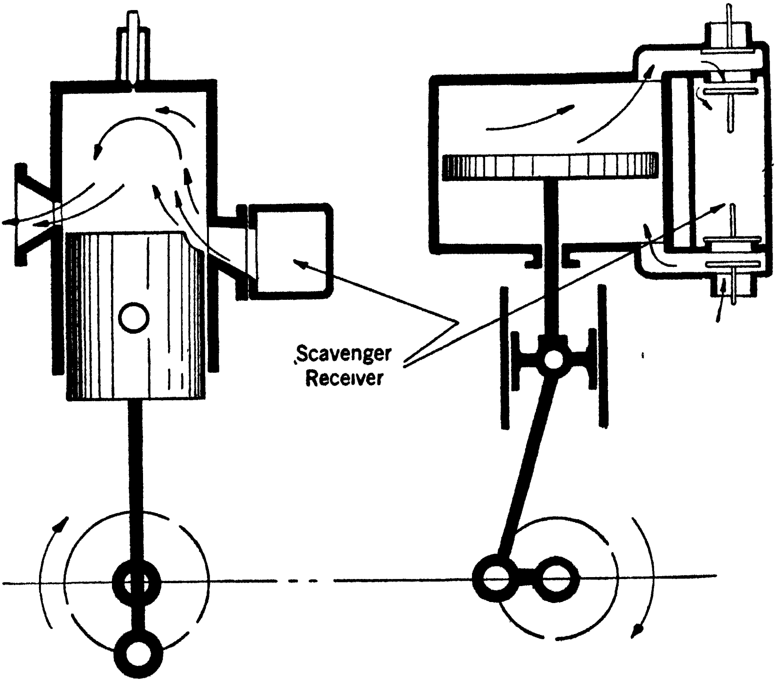


Fig. 10.—Simple 2-Cycle Port Scavenging Diesel Engine

by an air pump driven from the main crankshaft, is admitted by means of ports in the cylinder which are opposite a row of similar exhaust ports. As the piston descends, the exhaust port is first uncovered and the exhaust gases escape. As the piston further descends the row of scavenging ports is uncovered and scavenging air from the scavenging pump is blown into the cylinder. This air is at from 6 to 8 pounds pressure

and is deflected upwards by the deflector on the piston and by the angle of the scavenging ports. This air forces out most of the remaining gases and on the return of the piston the scavenging ports are again covered, after which the flow of scavenging air will stop. The piston then covers the exhaust ports and compression begins.

Overhead Valve Scavenging Type. With this type of 2-cycle engine the scavenging air is admitted by means of from one to four valves in the cylinder head as shown in Fig. 11. The action is as follows:

(A) Fuel is being injected into the cylinder and the

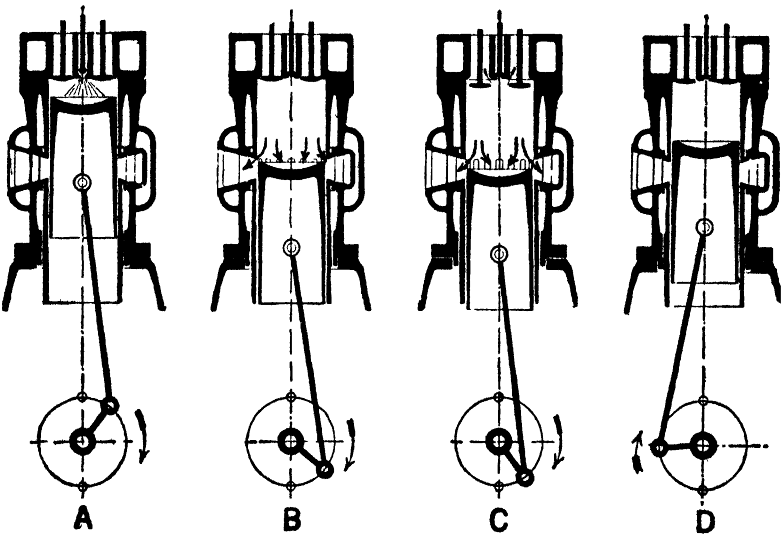


Fig. 11.—Overhead Valve Scavenging 2-Cycle Diesel Engine

piston is going down on its power stroke. All valves and ports are closed with the exception of the spray valve which remains open for about 30 degrees past top center.

(B) The piston starts to uncover the exhaust ports, reducing the pressure to atmospheric.

(C) An instant later the overhead scavenging valves open, and air, furnished by a separate air pump, enters and blows out the remaining gases through the exhaust ports.

(D) The piston is going up on its compression stroke. Shortly after covering the exhaust ports, the overhead scavenging valves close, leaving a charge of fresh air in the cylinder. This air is compressed as before and the cycle is repeated.

The overhead valve scavenging type of engine was formerly much used. It has the advantages of having the scavenging air enter at the top of the cylinder and giving better scavenging of the burnt gases, and permits the use of a flat top piston, which is an advantage. However, after many trials to produce a good engine having overhead scavenging, engine builders now admit that this is practically impossible because of troubles with the scavenge valves and with cracked cylinder heads and cylinders due to the excessive temperature changes caused by the alternate combustion of fuel and the expansion of the scavenging air through the scavenge valves, which causes a comparatively low temperature. These excessive variations in temperature subject the cylinder and cylinder head to great internal stresses which eventually result in cracked cylinders and heads.

Valve Controlled Port Scavenging Type. This type of engine combines many of the advantages and none of the disadvantages of the simple port scavenging and the overhead valve scavenging engines.

Simple port scavenging, while it overcomes the cracking of cylinder heads, possesses, in its ordinary form, characteristics which effect the engine detrimentally. It is obvious that the scavenging ports must not be uncovered by the piston until the pressure of the hot gases in the cylinder has fallen below the pressure of the scavenging air; serious scavenging air receiver explosions have resulted from insufficient attention to this matter of design. The terminal pressure of a Diesel engine is around 40 to 50 pounds pressure per square inch and the scavenging air pressure rarely exceeds 8 pounds. It is necessary, therefore, that the exhaust ports be uncovered in advance of the scavenging ports, and the amount of this earlier opening is usually about 8 per cent. of the stroke.

Uncovering the scavenging ports after the exhaust ports, naturally involves covering the exhaust ports after the scav-

enging ports; the result of which is that, at the time the upwards traveling piston covers the exhaust port, at about 20 per cent. of the stroke, the cylinder is filled with air at very little above atmospheric pressure, which is compressed during the remainder of the stroke. Thus, the weight of air compressed by this system does not exceed 85 per cent. of that of a cylinder full at atmospheric pressure.

Moreover, the scavenging by this method is imperfect, and there is an opportunity for burnt gases to blow back into the cylinder before the exhaust ports are closed. The cylinder therefore contains somewhat impure air. In very small engines the scavenging is improved by providing the piston with a deflector which guides the stream of scavenging air upwards; but in large engines a piston of this type would not last many days.

The engine shown in Fig. 12 utilizes port-scavenging, but employs two tiers, instead of only one tier, of ports. The piston uncovers the upper tier of scavenging ports before, and the lower tier after, it uncovers the exhaust ports, but the communication between the cylinder and the scavenging air supply or receiver, through the upper ports, is controlled by a timed and mechanically operated rotary valve, which remains closed until the exhaust ports have been uncovered long enough to reduce the pressure of the gases in the cylinder to nearly atmospheric; after which this valve is opened, while the piston uncovers the lower scavenging ports.

Upon its return stroke, the piston first covers the lower scavenging ports, and then the exhaust ports; the upper scavenging ports and their valve remain open, enabling the scavenging air to fill the cylinder at full scavenging pressure before the communication is shut off by the piston.

By referring to Fig. 12 the action will be seen as follows:

(A) During the first part of this stroke the spray valve is opened and fuel injection, combustion and expansion take place. All ports are covered by the piston and the scavenging valve is closed.

(B) The piston has uncovered the upper scavenging ports, but the scavenging valve remains closed. It has also started

to uncover the exhaust ports, allowing the exhaust gases to escape, and reducing the pressure to atmospheric.

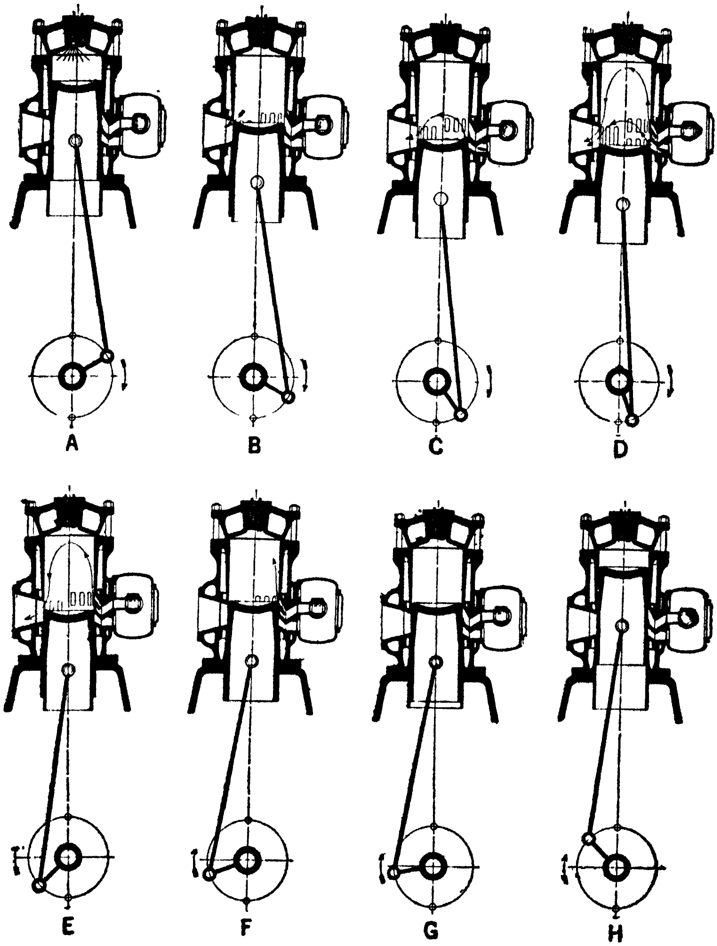


Fig. 12.—Valve Events of Busch-Sulzer, Valve Controlled, Port Scavenging, 2-Cycle Diesel Engine

(C) The piston is uncovering the lower scavenging ports and the scavenging valve begins to open. Air under 5 to 7 pounds pressure enters through the scavenging ports and blows the remaining burnt gases out through the exhaust ports.

(D) The piston has uncovered all ports fully and scavenging proceeds.

(E) The piston has covered the lower scavenging ports. The exhaust ports remain partially uncovered and the upper scavenging ports and valve remain open. Scavenging is almost completed.

(F) The piston has covered the exhaust ports but the upper scavenging port remains uncovered. The scavenging valve is still open and the cylinder is being charged with pure air at a low pressure.

(G) The piston has covered all ports and the scavenging valve begins to close. The piston starts to compress the air in the cylinder.

(H) The piston approaches top dead center and the scavenging valve is slowly closing. A few degrees before top center the spray valve will start to open.

Opposed Piston Engines. This type is known as the "Junkers" Type and is built to operate on both the 2 and 4-cycle principle. The 2-cycle type is shown diagrammatically in Fig. 13 and 14. In Fig. 13 both working pistons, A and B, are shown at their inner dead center. Upon the injection of fuel by the spray valve, V, the resulting burning and expansion of fuel drives both pistons outwardly. The lower piston, B, drives the crankshaft in the usual manner through a connecting rod, while the upper piston, A, is attached by means of a beam to two long connecting rods, each driving a crank at 180 degrees from the one driven by the piston B.

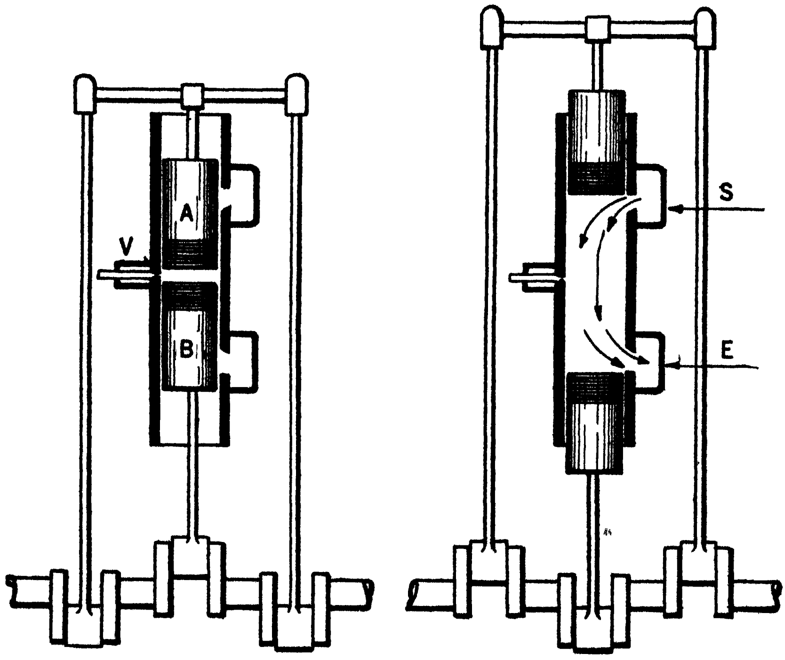
Both the top and bottom of the cylinder are open and the only valves required are one or two spray valves which are placed horizontally.

As the pistons reach the outer ends of their stroke, the lower piston uncovers an exhaust port, E, as shown in Fig. 14. An instant later the upper piston uncovers a scavenging port, S, and a current of scavenging air enters and sweeps the exhaust gases out through the exhaust port, and leaves a charge of fresh air in the cylinder for the next compression stroke.

The reciprocating masses of this type of engine are in perfect balance, and the reactions of the piston loads are taken

up by the moving parts instead of by the engine structure. Perfect scavenging is also obtained as the air enters at the top and has an unobstructed flow through the cylinder.

Double Acting Diesel Engines. A number of manufacturers are now building engines in which the power impulse is obtained during each piston stroke. The principle of a 2-cycle engine of this type is shown in Fig. 15. The exhaust manifold, E, has two sets of exhaust ports leading from the cylinder, each set of which is alternately uncovered by the ends of



Figs. 13 and 14.—Diagrammatic Arrangement of "Junkers" Opposed Piston, 2-Cycle Diesel Engine

the piston. Opposite the exhaust ports are two sets of scavenging ports leading from the scavenger receiver, R. The scavenging pump, P, supplies air at a low pressure to the scavenging receiver and is driven directly from the crankshaft.

In the position shown in the sketch, the working piston has just uncovered the exhaust and scavenging ports on its out stroke. Exhaust and scavenge is taking place at this end of

the cylinder, while at the other end the piston has just completed compression and fuel is being injected. On the in-stroke of the piston, combustion and expansion will take place until the piston uncovers the other set of ports. It will thus be seen that there is a power impulse at each stroke of the piston.

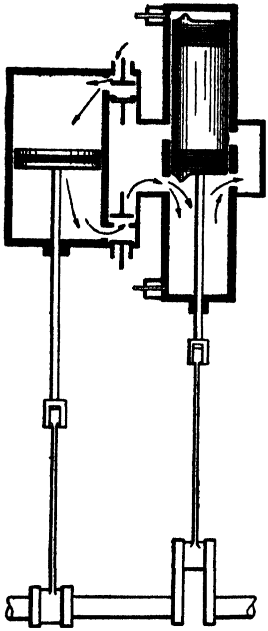


Fig. 15.—Diagrammatic Arrangement of Double-Acting, 2-Cycle Diesel Engine

Surface Ignition Oil Engines. These are commonly called semi-Diesel and hot-bulb engines and they operate on both the 2-cycle and 4-cycle principle. They differ both mechanically and thermodynamically from the Diesel engine. With this type the fuel is sprayed against a highly heated surface in a chamber having communication with the working cylinder. Contact with this highly heated surface ignites the fuel, which burns with explosion-like rapidity. They are quite often called "explosion oil engines," or engines in which the fuel is burned at constant volume.

Fig. 16 shows the principle of the 4-cycle surface ignition oil engine. While the cylinder is entirely water cooled, the

hot bulb, 1, is not cooled. When starting the engine, the bulb is heated by a blow torch to a dull red heat in order to ignite

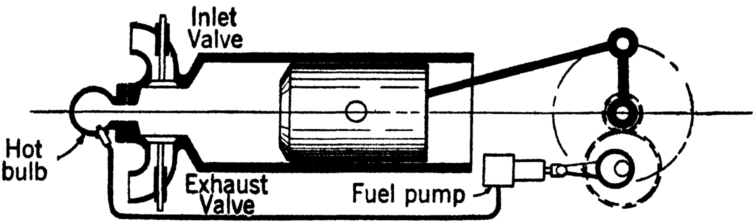


Fig. 16.—Diagrammatic Arrangement of 4-Cycle, Surface Ignition Oil Engine

the fuel sprayed into it by means of the fuel pump, 2. After the engine starts to fire, the blow torches are turned off, and the heat of the burning fuel thereafter keeps the bulb hot.

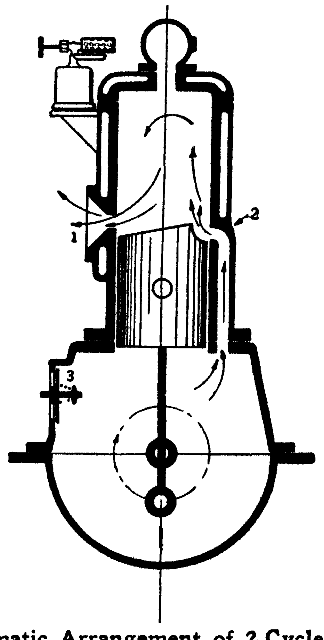


Fig. 17.—Diagrammatic Arrangement of 2-Cycle, Surface Ignition Oil Engine

On the out stroke of the piston, air is drawn into the cylinder through the inlet valve, 3. On the in stroke this air is com-

pressed to about 250 pounds per square inch and slightly before top center the fuel pump injects a charge of fuel into the hot bulb where it ignites and expands, forcing out the piston on its working stroke. On the next in stroke the exhaust valve, 4, is opened and the burnt products of combustion escape.

Most surface ignition oil engines will be found to operate on the 2-cycle principle of which the general mode of operation is as follows:

Referring to Fig. 17, suppose the piston to be on its downward stroke after the ignition. It will first uncover the exhaust port, 1, and the pressure will be released. As the piston goes further down it uncovers the inlet port, 2, and a charge of fresh air, previously drawn into the air-tight crank-case through the suction valve, 3, on the upstroke of the piston and compressed on the down-stroke, passes through the passage leading from the crank-case into the cylinder. This air scavenges out the burnt gases and leaves a fresh charge of air in the cylinder. As the piston again goes up this air is compressed to from 200 to 250 pounds pressure and the injection and expansion of the fuel takes place in the same manner as in the 4-cycle type.

For small powers a highly successful type of engine has been developed, in which the fuel charge is deposited during the suction stroke in a small perforated vaporizer extending slightly into the cylinder from the cylinder head. During the compression stroke a part of the charge is vaporized, and is ignited at the termination of the stroke. This results in an explosion in the vaporizer, which forces the remainder of the fuel through the perforations and into the cylinder in which the fuel is completely burned.

CHAPTER III

Comparative Efficiencies

Commercial Value of the Diesel Engine—Comparison of Two and Four-Cycle Types—Fuel Economy of Various Types of Engines—Comparison with Steam Plants—Advantages and Disadvantages—Availability of Power—Marine Diesel Engines—High Speed vs. Low Speed Engines—Diesel-Electric Drive.

The high commercial value of the Diesel engine lies in its unsurpassed fuel economy. This saving of fuel is not restricted to only the units of medium, and large sizes, but is also found to be almost proportionally as great in units of the smallest size. The engine has been developed to such a degree of perfection during the past few years that carefully built and designed engines can be considered to be fully as reliable as steam equipment.

Comparison of 2-Cycle and 4-Cycle Engines. For a given cylinder size and speed the 2-cycle engine appears to perform twice the work of the 4-cycle engine, since the former has a power impulse every revolution against one in every two revolutions of the 4-cycle type. The mechanical efficiency of the 2-cycle engine would also appear greater, as the 4-cycle engine has to make two strokes in expelling the burnt gases of combustion and charging the cylinder with fresh air for burning the next charge. The 4-cycle engine is required to have a much heavier flywheel from which to draw its stored up energy during the powerless strokes of the piston, and this energy has to be returned during the power strokes.

In actual practice, however, these advantages are not so apparent. The 2-cycle engine has the disadvantage of requiring a low pressure scavenging pump, which performs the work of scavenging and charging of the cylinders which, with the 4-cycle type, is accomplished directly in the cylinder on the exhaust and suction strokes of the piston. To insure thorough scavenging of the 2-cycle cylinder, a scavenging pump of large capacity is necessary, and much of this scaveng-

ing air is lost by passing out through the exhaust ports with the exhaust gases.

The fuel consumption of the 2-cycle engine is slightly greater than the 4-cycle engine, chiefly on account of the successive scavenging and charging of the latter which is more complete. The conversion of the fuel into useful work and distribution of the different heat losses of the two types of engines is shown diagrammatically in Fig. 18.

The 2-cycle engine has proved much more difficult to design, and the scavenging of the cylinders has to be solved mostly by costly experiment. If the scavenging is not perfect

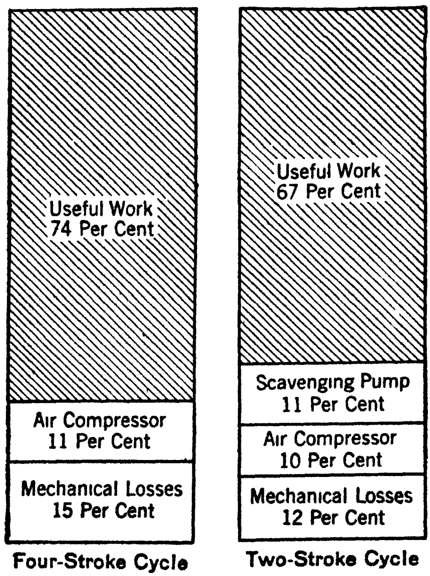


Fig. 18.—Diagram Showing Amount of Useful Work and Distribution of Heat Losses on 2- and 4-Cycle Types

the working of the engine will prove defective. Instead of having a cylinder full of air for combustion, with imperfect scavenging there remains a quantity of burnt gases which reduce the efficiency of the engine by not properly utilizing all the fuel.

Engines of the 4-cycle type have been very highly developed and their field lies in powers between 50 and 3,000

horse power mostly. The 2-cycle engine has lately been coming into the field for the larger sized units, and this type may predominate for the larger ones.

At the present time 2-cycle engines of as high as 800 horse power per cylinder are being built, and 4-cycle engines up to 1,000 horse power per cylinder and larger. Single cylinder engines developing 2,000 horse power have been built for experimental purposes.

Comparison of Diesel and Turbine Plants. The following is a comparison of the advantages and disadvantages of Diesel engines and the next most efficient power, the steam turbine.

Advantages of Diesel Engine Plants

1. Very high thermal efficiency and absolute heat rate, independent of the size of the unit.
2. Absence of all boilers, soot and smoke.
3. Constant readiness for operation.
4. No fuel losses in firing up and in unconsumed coal in ash as in steam plants.
5. Small volume of cooling water required.
6. Use of cheap liquid fuels, such as petroleum residues, coal-tar oils, and certain coal tars.

Disadvantages of Diesel Engine Plants

1. Limited size of units.
2. Massive foundations necessary.
3. High consumption of lubricating oil.
4. High installation costs.
5. High upkeep costs.

Advantages of Steam Turbine Plants

1. Unlimited sizes of generating units.
2. High rotative speed, reducing size of generator units, and cheapening generator construction.
3. Low installation costs.
4. High overloading capacity.
5. Ability to use under the boilers all classes of fuels, including the cheaper varieties of inferior grade coals.

Disadvantages of Steam Turbine Plants

1. The fuel economy is greatly dependent on the skill of the operating force and the degree of thoroughness with which the installation is maintained at operating efficiency.

2. Dangers and drawbacks of boiler operation.

3. Boiler losses generally and fuel losses in firing up and in banking boilers and in ash and clinkers.

4. Exacting requirements relating to circulating water for condensing the steam, one of the deciding factors in the commercial success of a steam plant.

A comparison of efficiencies of a turbine and Diesel plant

	Steam Turbines		Diesel Engines		
Load, Per Cent of Rated H. P.	Rating, 500 K. W Boiler Pressure, 150 Lbs. Vacuum, 26 Inches. Steam Per K. W. Hour, 21 Lbs.	Rating, 5,000 K. W Boiler Pressure, 200 Lbs Superheat 150° F. Vacuum, 28 Inches. Steam Per K. W. Hour, 14 Lbs	Engine Rating, 165 B H P.	Engine Rating, 520 B H P	Engine Rating, 2,200 B. H. P.
100	21,000	15,000	9,000	8,400	8,000
75	23,500	17,500	9,500	8,900	8,500
50	27,000	20,500	10,800	9,800	9,000
25	36,000	28,000	15,400	13,000	12,000

Fig. 19.—Comparison of B.T.U.'s in Fuel Consumed per B.H.P. Hour Between Diesel and Steam Turbine Plants

in regard to the B.T.U.'s in fuel consumed per brake horse power hour is shown in the table, Fig. 19. The curves in Fig. 20 show graphically the average consumption of heat units in fuel, in ordinary land power plants; A, being a simple, non-condensing Corliss engine plant; B, a compound, condensing Corliss plant; C, a steam turbine plant; and D, a Diesel engine plant. The flat curve of the Diesel, showing the main-

tenance of high efficiency throughout its load range, is characteristic of this type of prime mover.

Self Contained Prime Mover. The Diesel engine possesses the further great advantage of being a self-contained prime mover, its only auxiliary being the air compressor, which in nearly all engines is driven direct from the engine crankshaft. If directly connected to a generator, it comprises with the switchboard equipment a complete electric generating station. Its space requirements are therefore small, and it can be placed in the basements of buildings and hotels.

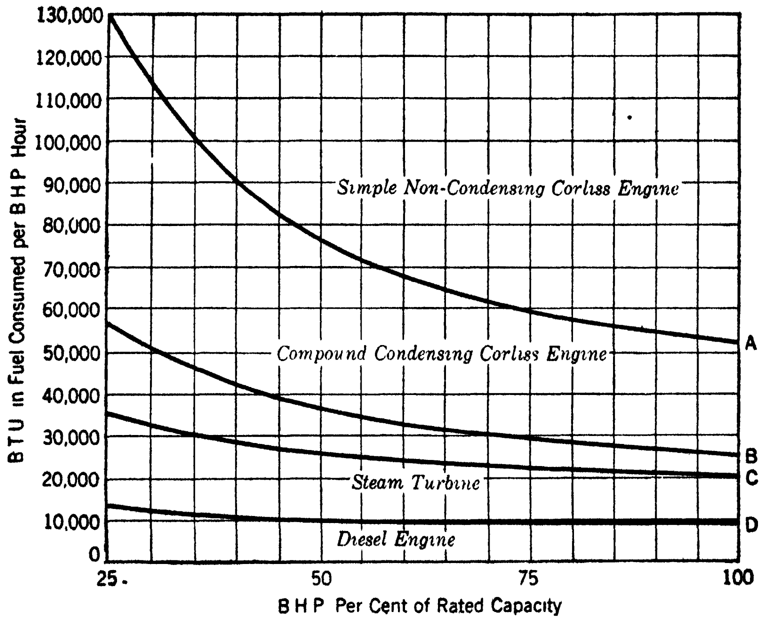


Fig. 20.—Curves Showing the Average Consumption of Heat Units in the Fuel in Ordinary Power Plants

Power Always Available. The engine is in readiness for instant operation and can take on full load from no load in an instant. With the stopping of the engine no further fuel is consumed. There are no stand-by losses, no losses in firing up or in banked fires, up the stack, in incompletely burned fuel and clinkered fuel, radiation and condensation due to inefficient firing or inefficient upkeep of the boilers.

The great economy of the Diesel engine makes even moderate sized central stations for small towns and isolated power plants feasible and renders possible the generation of such cheap power that the large central stations with overhead transmission systems can compete with them only in densely settled areas.

Most industrial plants that operate day and night, and therefore have an exceptionally high load factor, can generate their power with Diesel engines at such a cost, including all operating and fixed charges, that they can fix a profitable price that no central station can equal, even in powers as small as 100 horse power. Even with a 50 percent load factor, such an engine will, as a rule, generate power more cheaply than it can be purchased from a steam generator plant.

Marine Diesel Engines. The Diesel propelled motorship is the latest claimant for recognition in competition with the old established steam vessel, and, although the youngest in point of development, has shown in practice that the steam vessel has for the first time in its history a real rival that is demonstrating that steam is no longer supreme.

The superior economy of the marine Diesel engine over other types of prime movers is the cause. The fuel consumption of the Diesel engine averages about .39 to .42 pound per shaft horse power, while the very best performance with geared turbines and water tube boilers is about .90 pound per shaft horse power per hour. This is a low value, and the best that is usually obtained in practice is about 1 pound per shaft horse power day in and day out at sea. Comparing the Diesel engine with the most efficient steam turbine installation, it will be found that the steam installation has a fuel consumption of 2.5 to 3 times that of the Diesel engine. If the comparison is extended to coal burning ships with reciprocating engines the ratio is much greater.

For the cargo ship and small passenger liner, it is the ideal installation. Its adoption reduces the operating expenses on account of the reduced fuel costs and greatly increase the annual profits because of the greater carrying capacity brought about by the large reduction in fuel weight and space required for fuel bunkers. The weight and space occupied by fuel is

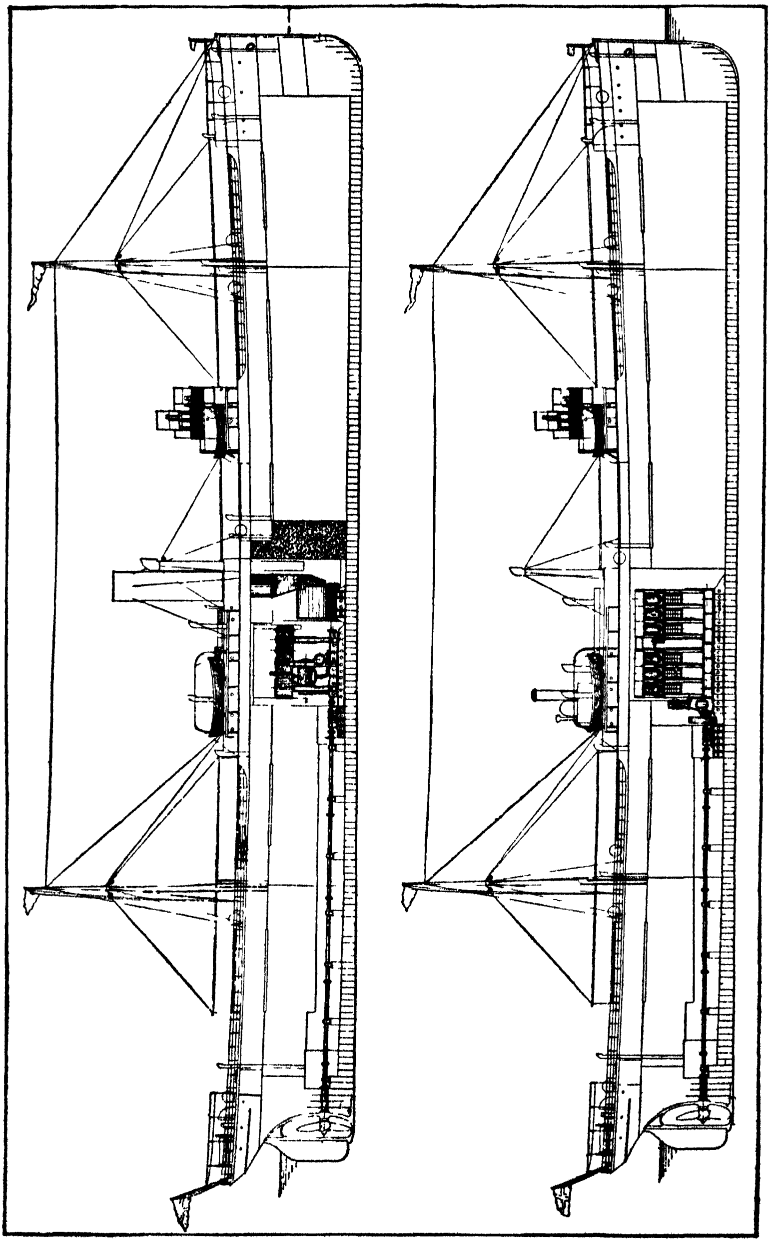


Fig. 21.—Comparison of Space Occupied Between Diesel and Steam Plants on Single Screw Ships

about 25 percent of that required for a coal burning, and 40 percent for an oil burning steamship.

Comparison of Diesel With Steam Machinery. Diesel machinery has the disadvantage when compared with steam, in being considerably heavier and has a higher first cost. It has the advantage in occupying less space, requiring a much smaller engine room force to operate and, as already pointed out, a fuel consumption of one-sixth to one-half of that of steam installations. The comparison of space occupied by the two forms of installations is shown in Fig. 21, both vessels having the same dimensions and speed.

The objections raised by vessel owners to the Diesel engine appear to be mostly imaginary—it is a new type of machinery still in a state of development and subjected to break downs; repairs may be difficult to obtain for this new type of machinery, causing interrupted service and long delays in port with consequent loss in profits; Diesel machinery is easily mishandled in the hands of an inexperienced crew; there is a scarcity of operating engineers and, if the Diesel engine is widely adopted, ships may be held up for lack of a trained engine room force; break-downs and troubles are frequently reported, and oil cannot be obtained in all ports.

Along with these objections it must be borne in mind that men trained in steam engineering hesitate to give up a well known and reliable type of machinery for a new and (to them) precarious type; and that new types of machinery have always been slowly adopted on shipboard as the operating engineer must be his own repairman and the engineer cannot call on the manufacturer for advice and assistance as could be done at a power plant on shore. This last is a strong argument against the rapid adoption of the Diesel engine for ship propulsion, and can only be overcome by well trained and competent operating engineers. Most manufacturers insist on a shop course and training of the men who are to operate their engines.

Two vs. Four Cycle Engines. There has been much discussion as to whether the 2 or 4-cycle engine is the better for marine use. Judged in the light of the number of motor

ships in successful operation, at present the 4-cycle has the advantage.

The 2-cycle engine is a far more attractive proposition on paper than the 4-cycle, but on the other hand is not so well

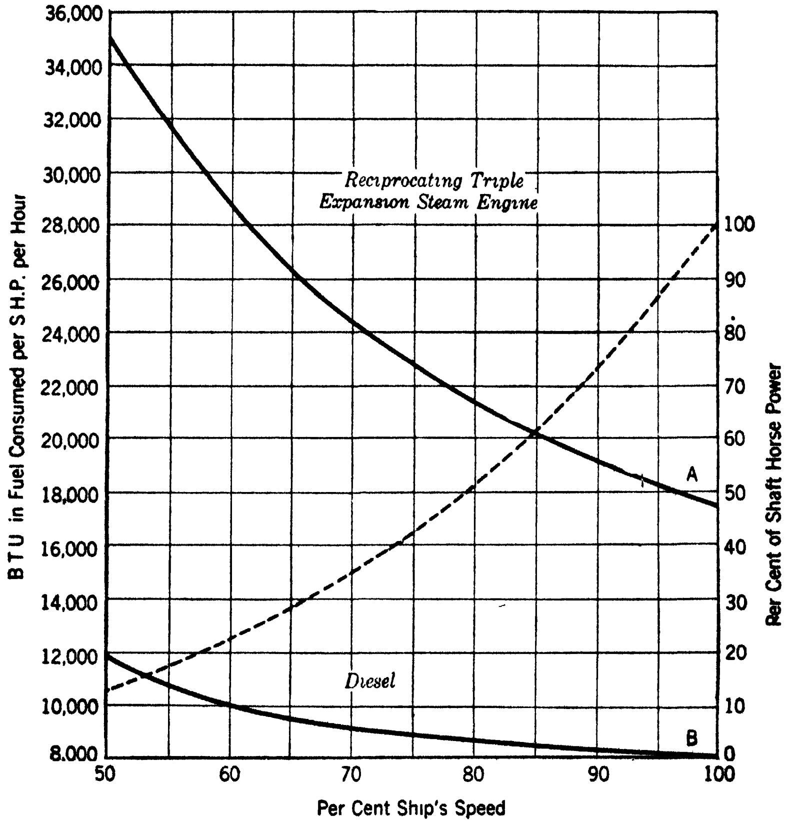


Fig. 22.—Curves Showing Relation Between Ship's Speed and Shaft Horsepower

known nor so universally used. It seems that the real question is not so much a comparison or selection of cycle as of make of engine. The 2-cycle engine is more difficult to design and is not so well understood as the 4-cycle engine, and there have

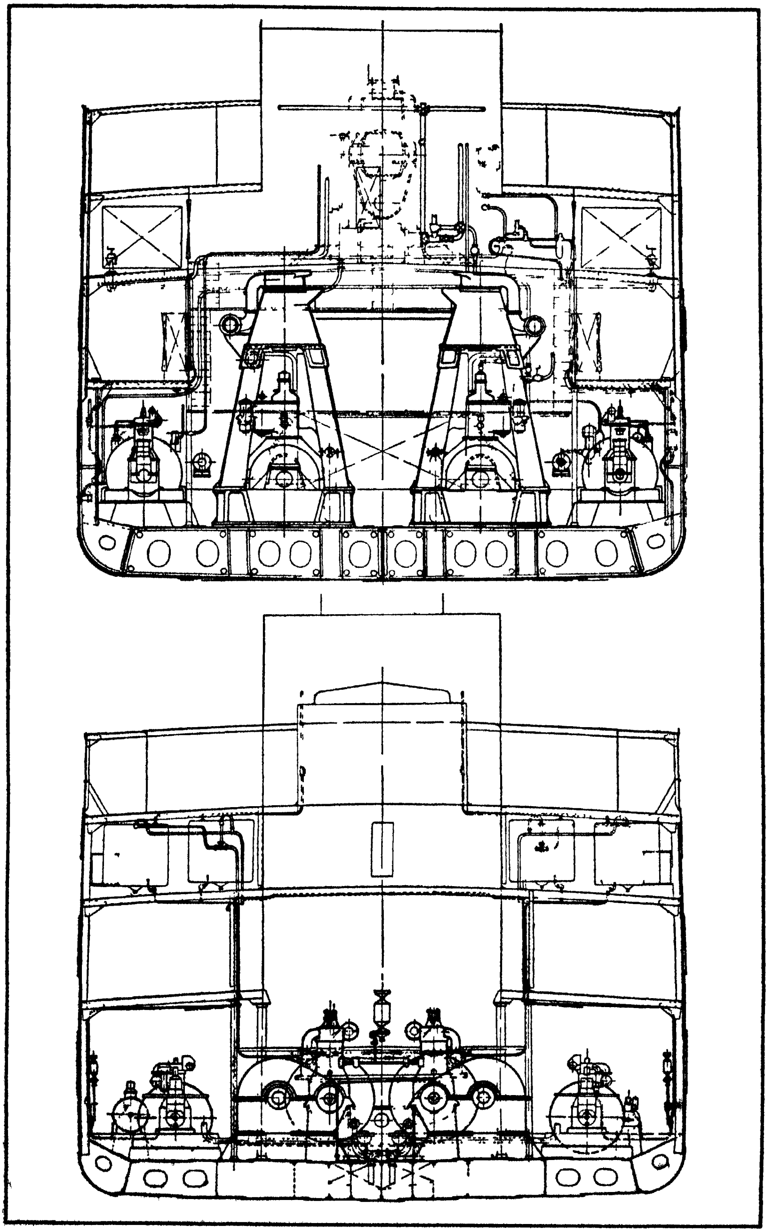


Fig. 23.—Section at Engine-room of Sister Ships, Motorship "Havelland" and Motorship "Westerland"

been more frequent breakdowns and troubles with this type of engine than with the 4-cycle type. Hence, those arguing for the 4-cycle engine have laid all these misfortunes at the door of the 2-cycle in general instead of the manufacturer in particular. Some builders of 2-cycle engines have had very good success and are making good most of the claims put forward for this type.

Many of the features of the 2-cycle engine are advantageous for passenger ships and ships where weight of machinery is an important item. The 2-cycle port scavenging type seems destined to have a large field in the future, especially after further development and experience.

In marine installations the power required to propel the ship diminishes much more rapidly than the ship's speed. The flat curve of fuel consumption of the Diesel engine is, therefore, of enhanced value. In Fig. 22 the dotted curve shows the relation between the ship's speed and the shaft horse power required to propel it, both in percentages of the full speed and load. Curve A shows the consumption of heat units in fuel of a high grade triple expansion reciprocating engine, and curve B, that of a Diesel engine, with reference to the ship's speed in per cent. of full speed. It will be noted that, although the fuel consumption at full speed is about 2.2 times as high for the steam engine as for the Diesel, at three-quarters speed it is 2.6 times as high.

High Speed vs. Low Speed Engines. There has been considerable discussion over the relative merits of the small high speed Diesel engine and the large low speed engine for ship propulsion. A great saving in space and weight results from the installation of small high speed engines which is offset by the greater reliability of the larger low speed engine.

Fig. 23 shows a sectional drawing through the engine room of the motorship "Havelland", equipped with small, high speed, 4-cycle engines with reduction gears, and its sister motorship "Westerland", equipped with slow speed, direct driven, 2-cycle engines.

Diesel Electric Drive. The merits of this type of installation are that a number of small high speed Diesel engines, driving electric generators, can be installed and the ship pro-

pelled by slow speed electric motors and hence high propeller efficiency obtained without reducing the engine revolutions. For the electric drive the first cost will be higher than for direct Diesel drive; the saving in weight between light high speed engines, generators and motors, compared with a large slow speed engine is small, and the installation is made much more complicated.

CHAPTER IV

Details of Construction

Crankshafts—Angularity of Cranks—Crankshaft Balance—Crankshaft Failures—Detecting Flaws in Crankshafts—Scored Crankshafts—Bedplates — Housings — A-frame Construction — Cross-head Type Framing—Submarine Engine Construction—Main Bearings—Babbitt Bearings—Fitting Bearings—Bridge Gauges—Cylinders—Relief Valves—Cracked Cylinders and Heads—Warped Cylinders—Gaskets and Joints—Cylinder Heads of Various Types—Pistons, Cooled and Non-Cooled Types—Piston Rings—Wrist Pins—Piston Seizures—Cracked Piston—Connecting Rods—Exhaust Valves—Inlet Valves—Piping—Valve Timing—Leaky Valves—Valve Troubles—Suction Strainers—Spray Air Compressor—Coolers—Foundations—Erection —Exhaust Heated Evaporators—Mufflers and Silencers.

Crankshafts. The crankshaft is the most expensive single part of an engine and is the most important and largest moving part. They are usually made of high-carbon steel or open hearth steel having a tensile strength of 30 to 35 tons per square inch. Light, high speed marine engines usually have crankshafts made of chrome-vanadium or other alloy steel of high tenacity.

The majority of Diesel engines having four cylinders or less have one-piece crankshafts turned from solid forgings. Engines having more than four cylinders usually have crankshafts made in two or more sections. This makes them easier to renew and less expensive to manufacture.

Figure 24 shows a typical marine, two section crankshaft for a six cylinder, 4-cycle engine. The extra crank at the forward end is for driving the spray air compressor. As common with most marine Diesel engines, the crankshaft is bored hollow for lightness and for the passage of lubricating oil to the crank journals. By removing its central portion the weight is reduced about 20 per cent. and the strength only a slight amount.

Angularity of Cranks. The angularity of cranks depends on the number of cylinders and the cycle of the engine. The

object is to equalize the period of working strokes and thus obtain an even turning moment. Four-cycle engines have the cranks placed at 360 degrees for two, 240 degrees for three, 180 degrees for four, 144 degrees for five, 120 degrees for six,

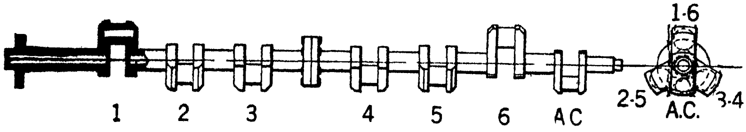


Fig. 24.—Two-section Crankshaft for Six Cylinder 4-Cycle Marine Diesel Engine

and 90 degrees for eight cylinders. With 2-cycle engines the cranks are placed at 180 degrees for two, 120 degrees for three, 90 degrees for four, 72 degrees for five, and 60 degrees for six cylinders.

In a 4-cycle engine, each cylinder has a power stroke every other revolution of the crankshaft, and with an engine having several cylinders, no matter how the cylinders are arranged to divide the power equally, it will not be possible to obtain more

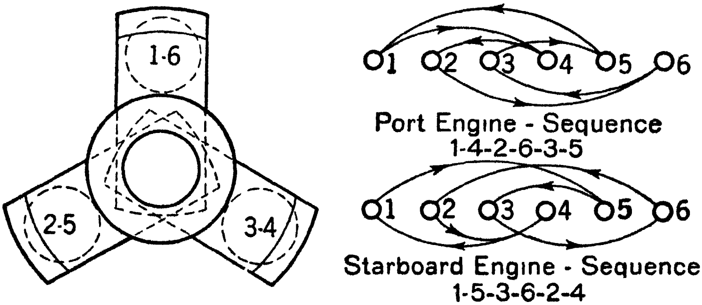


Fig. 25.—Arrangement of Cranks and Firing Sequence of Six-Cylinder, 4-Cycle Marine Engine Crankshaft

than one-half as many power strokes as there are cylinders. With 4-cycle engines it is common practice to arrange the cranks in pairs as shown in Fig. 25. It will be seen that cranks No. 1 and 6 are in the same position and their cylinders will fire just one revolution apart. Cranks No. 2 and 5 are also arranged together and their cylinders will fire one revolution apart, but one-third, or 120 degrees later than cranks 2 and 5.

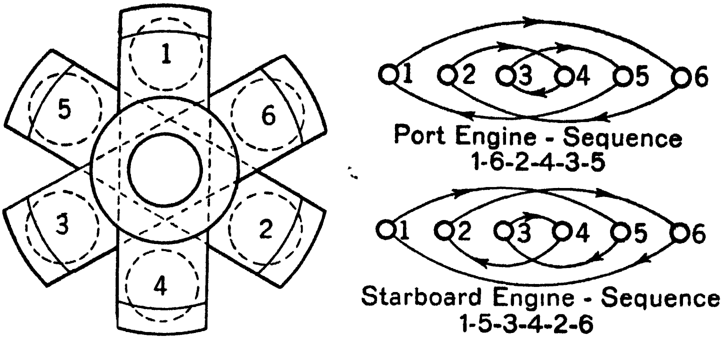


Fig. 26.—Arrangement of Cranks and Firing Sequence of Six-Cylinder, 2-Cycle Marine Engine Crankshaft

It will be seen that there will be three power strokes each revolution, and they will occur 120 degrees apart.

Two-cycle engines have a power stroke each revolution

NO. OF CYLINDERS	ARRANGEMENT OF CRANKS	FIRING ORDER
4 CYCLE ENGINES		
2	1 2	1-2
3	1 2 3	1-3-2
4	1 2 3 4	1-2-4-3
6	1 2 3 4 5 6	1-5-3-6-2-4
8	1 2 3 4 5 6 7 8	1-6-2-8-4-7-3-5
2 CYCLE ENGINES		
3	1 2 3	1-2-3
4	1 2 3 4	1-4-2-3
6	1 2 3 4 5 6	1-4-5-2-3-6

Fig. 27.—Arrangement of Cranks and Firing Order of Different Types of Diesel Engines

for each cylinder. The cranks are arranged equally all the way around the circle as shown in Fig. 26, which is an end view and shows the firing sequence of a typical six cylinder, 2-cycle, marine Diesel engine crankshaft. The end views of

the shafts are shown as they appear when looked at from the after ends of the port and starboard engines, respectively.

The arrangement of cranks and sequence of ignitions as shown in Figs. 25 and 26 allow both the port and starboard engines to have interchangeable crankshafts, although they both necessarily run in opposite rotation. Fig. 27 shows the arrangement of cranks and firing order of different types of Diesel engines.

Crankshaft Balance. There are two forces acting on a crankshaft to unbalance it: Those forces of the reciprocating masses of the piston and connecting rod, and cross-head and piston rod, if so equipped, and the centrifugal forces due to the rotating masses of the crankshaft, fly-wheel and crank brasses.

The forces due to the reciprocating masses are usually left unbalanced. Frequently, however, the centrifugal forces are balanced by counterweights bolted to the crank webs opposite the crank journals, as shown in Fig. 28.

Crankshaft Failures. The failure of a crankshaft is so fatal that one is surprised that designers ever allowed it to

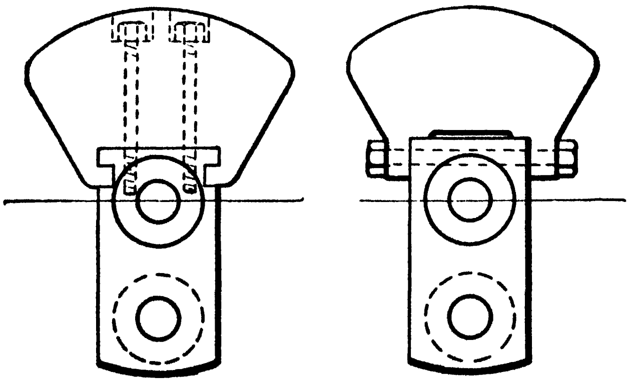


Fig. 28.—Common Methods of Securing Counterweights to Crank Webs

become a possibility, and yet in the early days, more than one engine failed in this item. By reason of its geometrical form, any crankshaft is a weak structure, and in a Diesel engine any faulty setting or operation of the spray valve may cause very heavy loads to come upon it, so a correctly designed crankshaft is to be desired, and will save unnatural strains coming on the engine structure.

Another cause of crankshaft failure is lack of alignment of the main bearings, resulting from unequal wear in the main bearings which support the shaft.

In such cases, almost without exception, the crankshafts fail through fracture of the webs. The cracks start at the center and gradually work out. Either web may fail in the single cylinder unit, but with two-cylinder units, web No. 4 fails most frequently. (The webs being numbered consecutively commencing at the end farthest removed from the fly wheel.) With three-cylinder units the fracture most commonly occurs in web 1 or 6, and with four-cylinder units it is almost invariable that web 4 fails.

Data was obtained from two engines having four cylinders each, extending over a period of several years. One broke its shaft about four years after its installation, the other in about five years. In both cases web No. 4 failed. New crankshafts were secured and after being bedded in, the engineer took out the lower halves of the bearings and measured down from the facings to the white metal. This was done by means of a gauge, described further on in this Chapter. Following this the bearings were measured once each year and the actual wear recorded. In one case for three and one-half years and the other for four years. The results showed that No. 3 main journal bearing experienced the greatest wear. The data obtained was plotted to scale with depths as ordinate and bearing numbers as abscissas. This gave a ready graphic view of the relative wear of all the bearings. The result obtained is characteristic of four crank engines, and since the shaft undergoes repeated shocks due to the reversal of the thrust, it will tend to bend in that section where the bearings are most worn. This explains why web No. 4 (the after web of No. 2 crank) fractures in the four-cylinder engine.

Still another cause of crankshaft failure is that due to the strains set up by torsional vibrations. In these cases the failure, generally speaking, takes place in one of the journals, which breaks at approximately 45 degrees and if there is an oil hole in the shaft, the break—or crack—will undoubtedly pass through the oil hole and other cracks will be there at right angles to it, also passing through the oil hole.

While a shaft may be correctly designed for a certain speed and method of drive, a change in either, from the original design, may be fatal so far as the shaft is concerned.

Detecting Flaws. Crankshafts of Diesel engines are subjected to great strains, and flaws often appear after short usage. Small cracks which develop should be carefully inspected for depth. A good method of telling whether a suspected crack is just a surface crack or if it is deep is to build up a dam of putty or red lead around it and fill with kerosene oil, which is allowed to stand for several hours. The putty and oil are then removed and the crack wiped thoroughly dry and polished with a clean rag. A white cigarette paper is then placed over the suspected crack and the journal given a heavy blow with a lead maul. If the crack is deep the paper will be well oil stained, but if only a surface crack no oil will show on the paper.

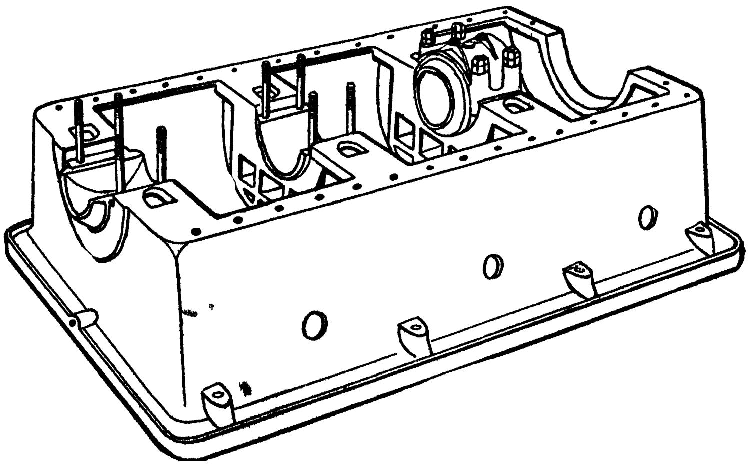


Fig. 29.—Bedplate of Small Stationary Engine

Scored Crankshafts. It sometimes happens when there has been trouble with wiped and burnt out bearings that the crankshaft journals become scored. These scorings should be carefully stoned with the aid of an oil stone, or, in careful hands, in the case of badly scored journals, with a fine mill file and finished by stoning. The grooves are not very dangerous if carefully smoothed down.

During overhauling, when the bearing caps are removed, the bearings and journals should be well covered with rags or canvas to prevent dirt and foreign bodies from falling into the bearings.

Bedplates. The bedplate of an engine may be called the backbone of the fixed parts of the engine, that is, it is the part which holds the entire engine and upon which the other parts are built. With marine engines, the bedplate generally rests upon heavy girders, or engine bearers, and these are in turn secured to the framing of the ship. With stationary engines

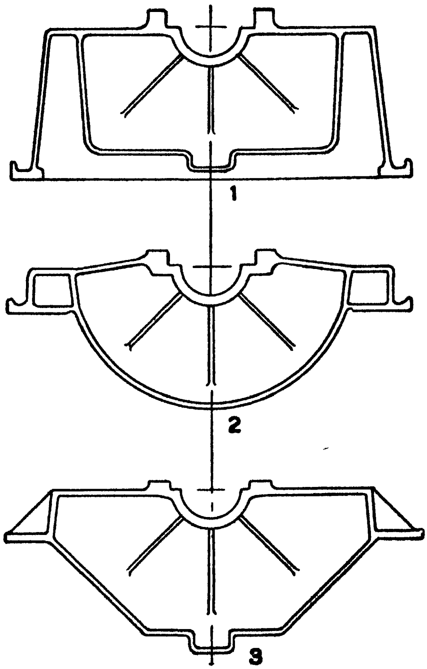


Fig. 30.—Common Forms of Bedplates

the bedplate is firmly set on a concrete foundation. Fig. 29 shows a typical bedplate for a small stationary engine. The bedplate has two important duties to perform. First, to carry the housing, or framing of the engine above, upon which the working cylinders are in turn carried, and second, to carry the main crankshaft bearings which in turn support the crankshaft.

The bedplate is generally a solid one-piece casting of cast iron. Marine engines sometimes have bedplates and housings of bronze or cast steel. Large engines having many cylinders usually have bedplates made in two or three sections and secured together with special body-bound bolts, or else with keys, in addition to the regular bolts. These are for the purpose of keeping the different sections of the bedplate in perfect alignment. The upper faces of the bedplate, together with the bearing saddles, are carefully machined in one setting on the bed of the planer during manufacture. The bearing saddles are usually cut by a special spiral tool, the radius of which is centered in the planer head.

Fig. 30 shows the shape of the most common types of bedplates. Type 1 is the most usual design for stationary engines, as it raises the center line of the crankshaft higher, enabling

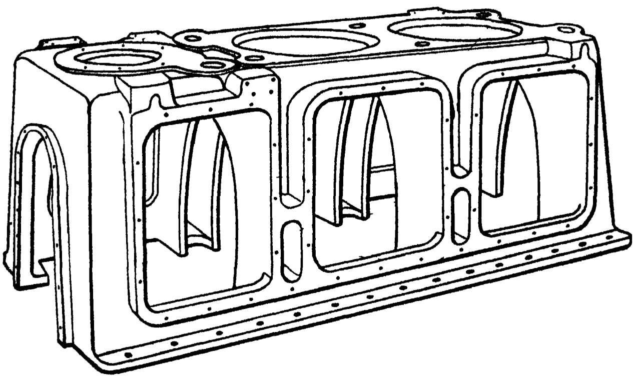


Fig. 31.—Box Type Housing

a generator to be erected flush with the floor or mounted on an attached sub-base. The one shown at 2 is also a stationary type, being usually designed for A-frame construction. Type 3 is a design associated with marine engines, as it gives a low crankshaft center line.

Housings. Housings, or crank cases, are used to support the working cylinders and provide space for the rapidly rotating crank shaft and connecting rods. The great majority of Diesel engines have housings similar to that shown in Fig. 31, which is known as a "box" housing. The large openings at

the side are for removable doors which allow inspection and adjustment of the main bearings and crank brasses. Engines having six or eight cylinders often have housings made in two or three sections secured together in a similar manner to the bedplates.

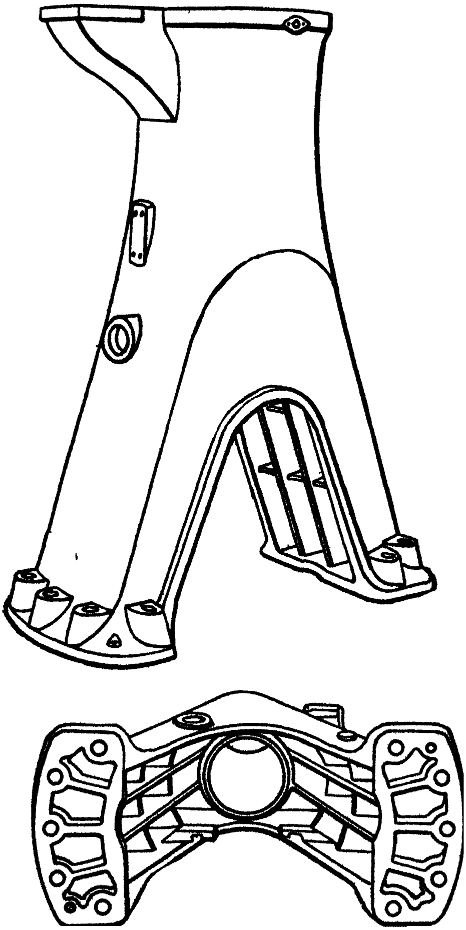


Fig. 32.—Frame Construction of Fulton-Tosi Stationary Diesel Engine

A-Frame Construction. The first Diesel engines made used this form of construction and on account of its merits is still very extensively used. Fig. 32 shows the A-frame of the Fulton-Tosi Diesel engine. A cylinder liner is fitted into the

upper part of the A-frame, from which it may be readily removed and replaced. The space between liner and frame is used as the cooling-water jacket.

Cross-Head Type Framing. Fig. 33 shows two forms of construction commonly found in large crosshead type Diesel engines. Type 1 has two cast standards similar to marine steam practice, with guides for the crosshead. Type 2 has one cast standard and a removable support of forged steel.

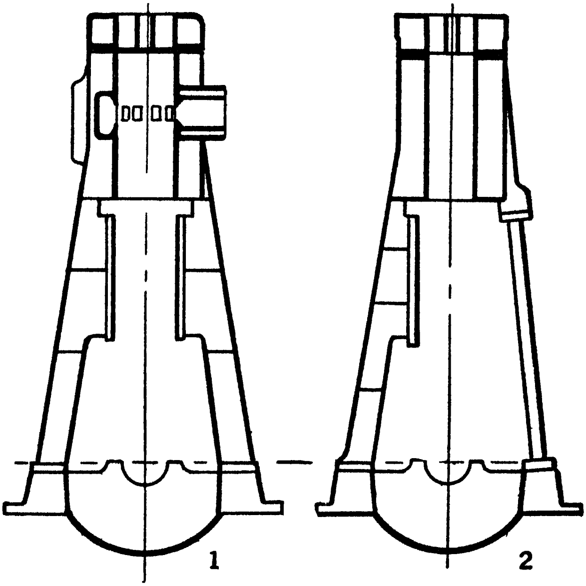


Fig. 33.—Typical Frame Construction of Cross-head Type Engines

This construction allows for ready removal of the crankshaft sideways from the engine, while with the former type the shaft must be removed endwise, which is sometimes very difficult in marine practice on account of limited space.

Scavenging Pumps. There are four general methods of supplying air for scavenging the working cylinders of the burnt products of combustion and for leaving a fresh charge of air in the working cylinders for the compression stroke.

The first and oldest method is by attaching the scavenging-pump piston directly to the working piston, each working

cylinder having its own scavenging pump. Such construction can be seen by referring to Figs. 180 and 184. The scavenging pump of the Southwark-Harris engine serves a double purpose as it is upon this piston that the starting air acts when starting up the engine by compressed air. The air intake into the scavenging cylinder is at A, which the scavenging piston uncovers when near its bottom stroke. On its upward stroke the air is discharged through the passage C, and discharge

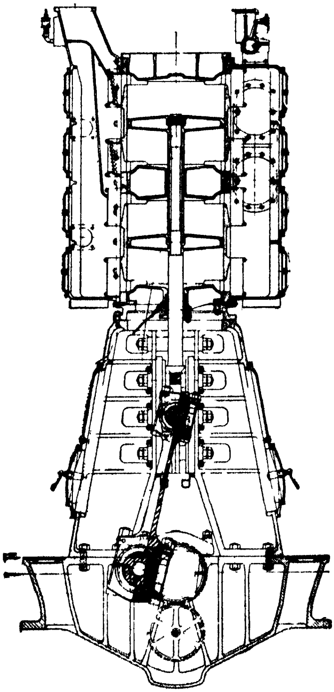


Fig. 34.—Cross-Section Through Scavenging Pump of Busch-Sulzer 2-Cycle Diesel Engine

valve B, into the space or receiver, D, where it remains until the working piston uncovers the exhaust and scavenging ports. The scavenging air then enters through the ports and clears the working cylinder of its burnt gases.

In Fig. 184 the arrangement is somewhat the same with the exception of the valves. Air is drawn in through the lower valve on the upward, or suction stroke of the piston, and dis-

charged through the upper valve into a receiver on the upstroke

The second method of driving the scavenger pump is by means of a beam from one of the working cylinder crossheads or connecting rods. The third method and one which is coming into prominent use with very large 2-cycle engines is by means of independently driven turbo-blowers.

The fourth, and most common arrangement, is to drive the scavenging pump directly from the crankshaft and have but one such pump for all working cylinders. Fig. 34 shows a

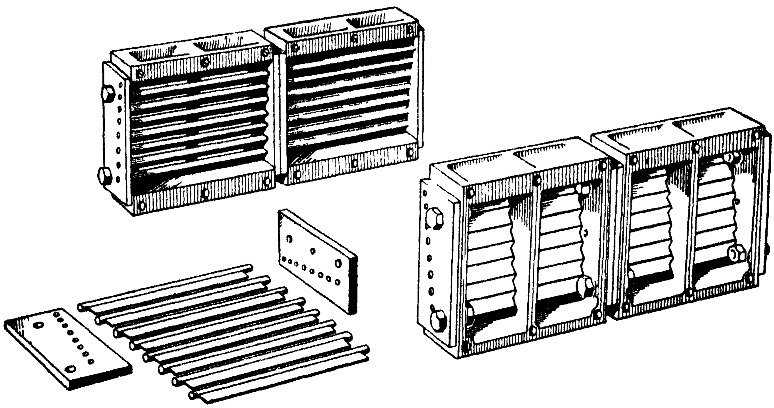


Fig. 35.—Automatic Shutter Valves of Busch-Sulzer Scavenging Pump

Busch-Sulzer scavenging pump directly driven by a crank on an extension of the main crankshaft and is provided with a crosshead and guide similar to those of the working cylinders. The suction and discharge valves, Fig. 35, are of a simple automatic "shutter" type, mounted in cages. The valves are identical in size and design and are interchangeable. The pump is of the double-acting type with tandem cylinders. The intake side of the pump is provided with a valve chest, so arranged that the scavenging air may be brought from outside the engine room. The discharge side is provided with a valve chest, with connections to the scavenging air receiver, which, in turn, provides the connections to the working cylinders.

Busch-Sulzer Submarine Engine Construction. The lightweight, high-speed submarine engines built by this firm are of

the general construction shown in Fig. 36. This shows the arrangement of the main parts of the crank case and bed plate, together with the lower end of a working cylinder, and the means of combining these parts.

In this figure, 1, is the working cylinder; 2, a cylinder support, and, 3, a tie-rod. The cylinder supports are of cast steel;

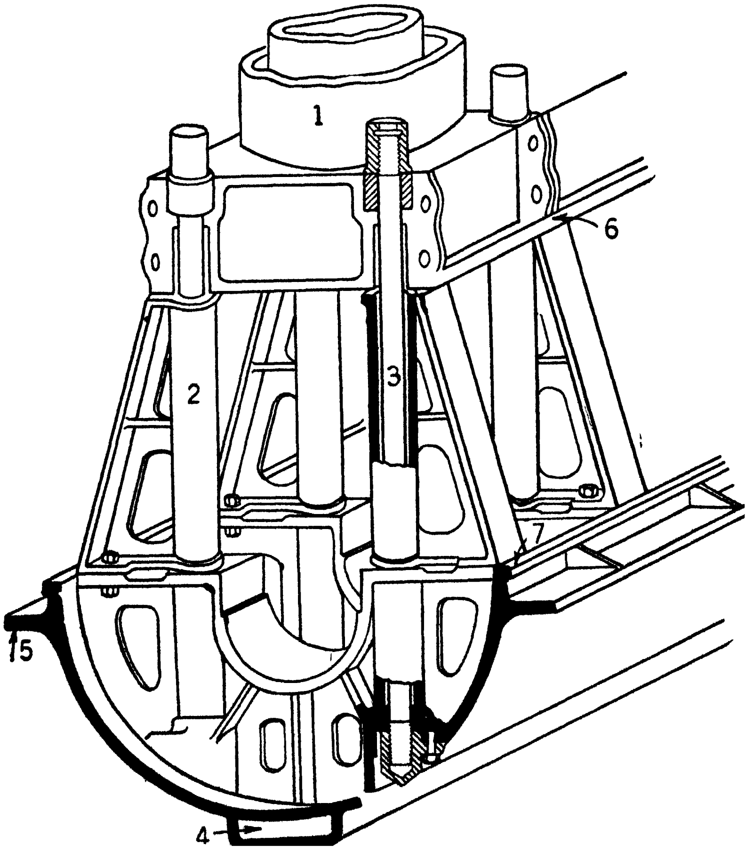


Fig. 36.—Construction of Busch-Sulzer, Lightweight Submarine Diesel Engines

the bed plate is a bronze casting. The tie-rods are forged steel rods.

The lower end of the working cylinders are spread out in the form of an entablature and the adjacent parts of these

entablatures butt together, with finished surfaces, thus forming the top of the crank case. The cylinder supports are located at the joints between the adjacent entablatures, at the corners of the latter, and a tie-rod passes through the joint at each support. Each tie-rod, therefore, serves two cylinders.

The tie-rod extends from the top of the cylinder entablature to the underside of the bed plate. At its top end it is provided with a special nut, resting upon a deep washer; the

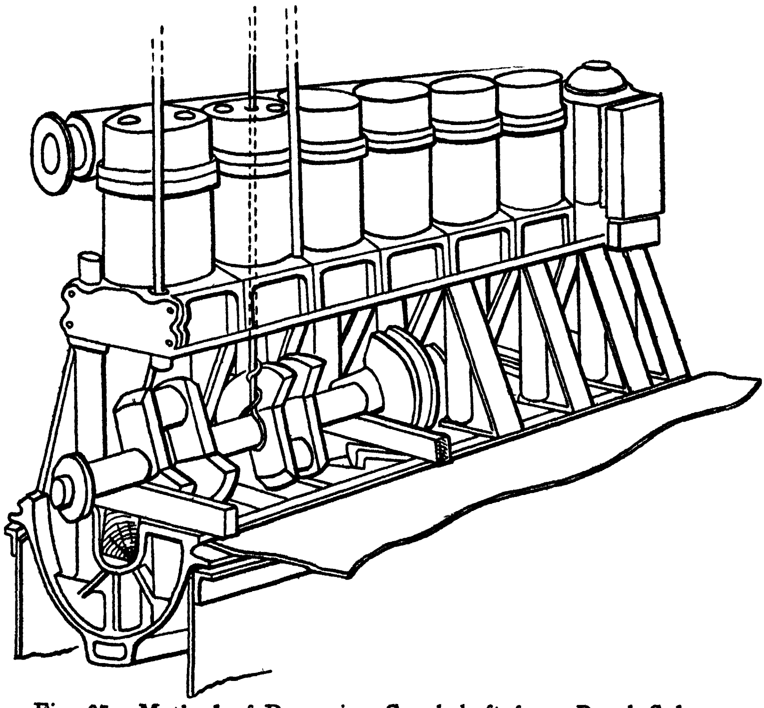


Fig. 37.—Method of Removing Crankshaft from Busch-Sulzer Lightweight Submarine Diesel Engines

latter fits into a recess, formed half in one entablature and half in the next. At its lower end, the tie-rod screws tightly into and bottoms in a bronze nut, attached to the bed plate, and this nut is provided with an extension, fitting tightly in a bored recess in the bed plate. The top end of the tie-rod and the upper nut are provided with a locking device; also with means for screwing the tie-rod into and out of the lower nut.

The correct alignment of the cylinders and bed plate, with reference to each other, is maintained by dowels and fitted bolts, in reamed holes, through the top and bottom flanges of the cylinder supports.

The bed plate is divided, in its length, into three main sections, which are bolted together by means of flanged joints. It is provided with seats for the main bearing shells. An oil channel, 4, extends along the bottom for draining off the lubricating oil. The flanges, 5, rest upon the engine foundation.

The cylinder supports are finished on their outer faces, as are also the edges, 6 and 7, of the cylinder and bed plate, for reception of the large sheet metal crank-case doors.

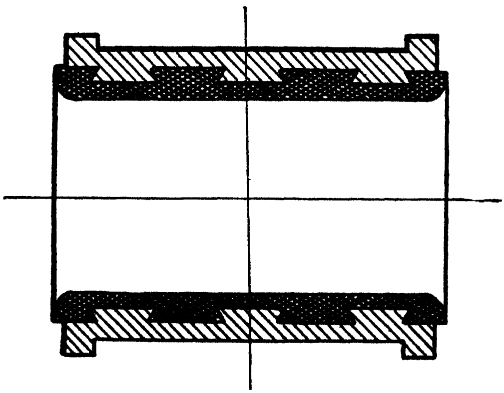


Fig. 38.—Method of Securing Babbitt Metal to Bearing Shells

One of the excellent features of construction of these engines is the ease with which a section of the crank shaft can be removed. It is not necessary to dismantle the cylinders, cylinder heads or valve gear. The cylinders can be hung up by their tie-rods and chain falls as shown in Fig. 37. The tie-rods on one side of the cylinders are withdrawn and the cylinder supports removed by sliding them outwards. The pistons and connecting rods can be removed from the bottom end of the cylinders and a section of the crank shaft lifted out as shown in the illustration.

Main Bearings. Main bearings are made in two halves, or shells, of cast iron, cast steel or bronze. The shells are lined with babbitt metal which is a composition consisting of about

80 per cent. tin, 15 per cent. antimony, and 5 per cent. copper. The bearings are first made of a much smaller bore than required by the crankshaft, hammered and peened and bored to exact size and hand scraped to a perfect fit on the crankshaft journals. The bearing shells are cast, having wedge-shaped depressions into which the bearing metal runs, thus giving the babbitt a strong grip to the bearing shells. This is shown in Fig. 38.

A novel form of main bearing construction is shown in the cross-sectional drawing of the main bearings of the Ingersoll-Rand, Price-Rathbun type engine, Fig. 38-A. It will be noted that the bearing caps, on which there is practically no wear, are bolted securely to the bedplate. The bottom halves of the

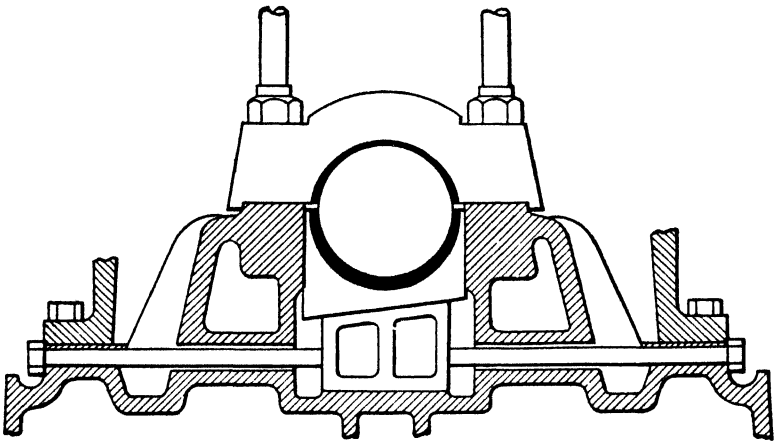


Fig. 38a.—Method of Adjusting Main Bearings of Ingersoll-Rand Oil Engine

main bearings are held rigidly sidewise in the bedplate and supported on heavy wedges which are adjustable from the outside of the housing by wedge bolts. By shifting the wedges the bottom half of each main bearing may be raised until the shaft is brought in contact with the rigidly held bearing caps, securing perfect alignment. The wedge bolts are then tightened and no further movement is possible. Likewise, by the removal of the wedge bolts and wedge, any one of the lower main bearings can be removed and renewed without disturbing the other parts of the engine.

Most horizontal Diesel engines have main bearings of the quarter-box design in either three or four parts. A typical design is that of the Allis-Chalmers main bearing shown in Fig. 39. The principal adjustment is made by the front wedge upon which part of the bearing the most wear occurs. The lubricating oil supply is of the stream type, supplied by a mechanical oil pump.

Babbitting and Fitting Main Bearings. The first step in babbitting a bearing is to melt out the old metal, preferably over a charcoal fire. The two shells are then thoroughly cleaned with acid, heated to a temperature of about 300 de-

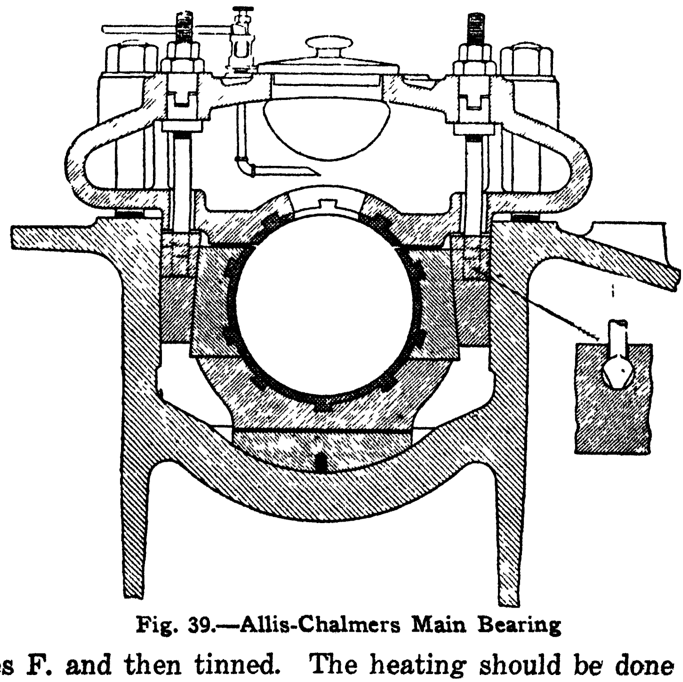


Fig. 39.—Allis-Chalmers Main Bearing

grees F. and then tinned. The heating should be done in an oven or over a gas or coke fire with the surface to which the babbitt clings turned upwards to prevent the formation of sooty and greasy deposits. A mandrel of a size suitable for the bearing should also be heated.

The two halves are bolted together with a shim of metal between the two shells to partly separate them. The bearing shells are then laid on their ends and the mandrel is inserted

in the center of the pair of shells; this mandrel representing the crank journal only being somewhat smaller. The setting up of the brasses preparatory to pouring the metal can best be solved by the machinist doing the job and with the materials at hand. All oil holes in the bearing should be plugged with clay or asbestos to prevent the escape of the babbitt while pouring.

Before pouring, care should be taken not to have the babbitt metal too hot, as if it is of too high a temperature, extreme shrinkage will occur, resulting in porous areas in the lining and in broken anchors; furthermore, the babbitt will become oxidized, softened and dirty. If it is poured at too low a temperature, a coarse granular formation is the result.

After pouring the metal into the brasses it is best to part the two halves before they have cooled off completely. The babbitt is hammered and peened to settle the metal and drive out all bubbles. The brasses are clamped together again, with the number of shims between them that were used on that particular bearing, and the bearing is then bored out to slightly smaller diameter than the crank journal to which it is to be fitted.

When a bearing jig is not at hand extreme care must be taken to bore the bearing out exactly central. A horizontal boring machine answers the purpose best. All bearings usually have a turned surface from which the axis of the boring bar can be determined. The face and fillet should be turned after boring.

The lower brass is then fitted to the main bearing saddle, and the latter scraped to a true fit with the under side of the lower shell. There should be a clearance of .002 to .003 inch between the lower brass guide flanges and the sides of the main bearing saddles; that is, just enough to permit the lower brass to move freely when it becomes necessary to roll the lower brass out of its saddle while the crankshaft is in place.

The lower brass is then scraped to fit the crank journal and set into place in its saddle. The crankshaft is dropped into place and the lower brass secured by dogs held by the bearing-cap holding down studs, and is thus prevented from turning.

The crankshaft journal is given a thin coating of Prussian blue or red lead, and rotated in its bearings. The bearing is then rolled out, or the crankshaft lifted, as the case may be, and the high spots of the bearings scraped. This is repeated until a perfect bearing surface is obtained.

When putting in or removing a crankshaft, great care must be taken to prevent straining and springing of the shaft by properly supporting it. When a crankshaft is being lifted by chain falls, the crank journals must be well protected by burlap or canvas.

After the brass has been scraped to fit the crankshaft journal the oil grooves are cut. These vary with different

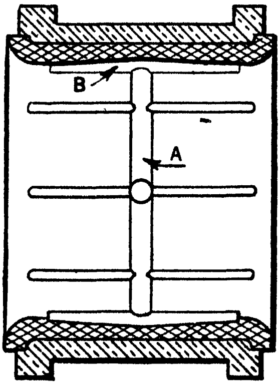


Fig. 40.—Oil Grooves in Main Bearings

makes of engines and it is a good policy to re-cut them as they originally were. A typical form of oil groove is shown in Fig. 40. The center groove, A, is somewhat deeper than the others as it carries oil around to the side clearances, or pockets, B. The small cross grooves are not carried all the way to the edges, as considerable oil would be lost if this was done. The edges of the grooves must be carefully dressed off. The side pockets, B, are cut on each side of both the upper and lower brasses and have a clearance of about .030 inch and a depth of about .75 inch for a six-inch shaft. They are usually tapered to each side as shown.

The main bearing cap is now scraped to a perfect fit to the crankshaft journal and is then set in place together with the

estimated number of shims to give the proper clearance. Lead wire, about No. 20 gauge, is placed in the upper brass near the two ends and in the middle. This wire can be conveniently held in place by a dab of Albany grease. The holding down nuts are then firmly tightened and then slacked away and the bearing cap removed. The leads are measured with a micrometer and the necessary further scraping, or addition and subtraction of shims, is estimated from these measurements.

Leads can not be depended on to give the absolute clearance, but are excellent in obtaining the relative clearance. When taking leads, the lead will compress only up to a certain

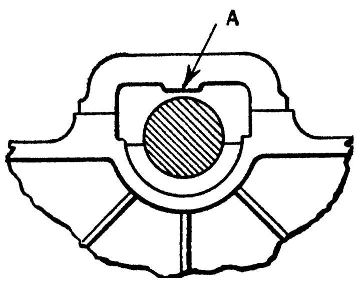


Fig. 41.—Bridge Gauge for Measuring to Crankshaft

point when the cap is pulled down, and any further pressure will only cause the lead to imbed itself into the soft metal of the bearing. If No. 20 gauge wire is used the running clearance will be right if the lead squeezes out to .008 inch, if the required clearance is between .006 and .008 inch.

Bridge Gauges. When the engines are built the manufacturer usually furnishes a bridge gauge, Fig. 41. This gauge is used to measure the amount that any bearing has dropped due to wear. To obtain a reading, the upper bearing cap is removed and the gauge set in position as shown. The distance from A to the crankshaft journal is measured by feelers and compared with the original measurements taken during the first erection.

Another, and probably a better method of obtaining the bearing wear is by the micrometer gauge shown in Fig. 42. Readings are taken at three points. Care must be taken to prevent loose white metal from falsifying the reading.

The careful engineer will keep a record of the amount of wear of the main bearings. An excellent way is to plot the bridge gauge readings to scale. The shaft can then be assured of perfect alignment by taking the most worn bearing as a reference point and the chart will show exactly how much must be scraped off the high bearings, or how much the low bearings must be shimmed up, to regain conditions of minimum stress on the shaft.

Cylinders. The cylinders of most large engines are provided with separate liners and heads. With small engines the cylinder and jacket, and sometimes the head, are cast in one piece. The material used is always close grained cast iron,

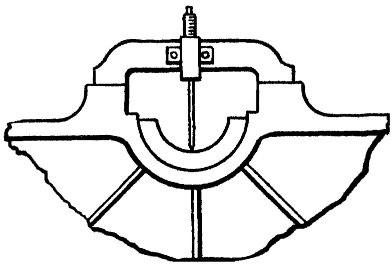


Fig. 42.—Bridge Gauge for Measuring to Bottom of Bearing Shell

although a number of engines have been built having cast steel jackets fitted with cast-iron liners.

Owing to the great differences in temperature between the cylinder and jacket and the different degrees of expansion and contraction between the two parts, the separate liner construction has proved more satisfactory. Such form of construction can be seen by referring to the Nordberg engine, Fig. 189 or the Busch-Sulzer engine, Fig. 183. The liners are rigidly held at the top end and are free to expand and contract longitudinally.

Fig. 195 shows the construction of the small Nelseco engines in which the cylinder, cylinder head, and jacket are cast in one piece. Such a design is only suitable for small engines up to about 30 horse power per cylinder.

Cylinder Relief Valves. In order to relieve any abnormal pressure which may be built up in a cylinder when an engine

is being started, due to an excessive supply of fuel oil, or to other causes, relief valves are fitted. These are ordinary spring-loaded relief valves, set to blow off at about 750 pounds pressure per square inch.

Cracked Cylinders and Heads. Cracked cylinders and cylinder heads are usually caused by unequal heating due either to poor design, poor quality of casting, air pockets, insufficient amount of cooling water or overloading. Cylinder explosions are also a frequent cause.

The first indication of a cracked cylinder or head will be made by a white steaming exhaust, and the particular cylinder affected may be located by opening the cylinder indicator cocks in turn until the one giving a steam vapor in the exhaust is found.

Troubles arising from air pockets and unequal heating due to air pockets are usually confined to the cylinder head. Pet cocks are most always provided to relieve the trapped air at the highest point on each cylinder and manifold. These pet cocks should always be opened when first starting up the engine to release any air trapped in the circulating water system, and again at intervals during operation to see that solid water is present in the jacket space. Where pronounced local hot spots occur, especially at a high part of the cylinder or head, a pet cock should be located to release the steam and air and allow better circulation of cooling water.

Cracked cylinder heads and liners, as well as cracked pistons, are due in many cases to overloading. These cracks often take place when the engine as a whole is not overloaded but is running light. The only explanation which seems to fit this condition is that of individual cylinder overloading. The obvious remedy is to take indicator cards at frequent intervals.

Cracked cylinder heads are sometimes caused by unequal expansion of the thin walls of uncooled metal between the recesses of the valve openings. The use of small core holes gives poor access to the interior of the cooling space for the purpose of cleaning away scale which gradually accumulates, preventing the proper cooling of the casting and resulting in unequal heat stresses.

The shut down of the circulating water pump for a few moments and then suddenly starting up again allowing cold circulating water to enter the jackets has caused many cracked cylinder and heads. For this reason most marine Diesel engines have separately driven circulating water pumps.

Warped Cylinders. Cylinders and pistons often become out of round due to casting stresses developed by overheating. Piston seizure is usually the result. The cylinders have to be rebored or fitted with new liners if the warpage is serious.

Cylinder Head Gaskets. The joint between the cylinder head and cylinder is usually made with special copper-asbestos gaskets furnished by the manufacturers. Sheet packing does not make a satisfactory gasket. Gaskets cut from 1/32 inch annealed sheet copper are used with excellent results.

Cylinder Heads. The most important elements which affect the reliability of a Diesel engine are the combustion space in the working cylinder and the parts surrounding this space.

The very high temperatures which occur in this space make it imperative that the surrounding parts be so constructed that excessive heat stresses will be avoided. This is a comparatively simple problem in the case of such parts as the cylinder liner and piston, as these are of symmetrical circular form, which will expand uniformly; but is much more difficult in the case of the cylinder heads, which must necessarily contain valve openings. If more than one or two valve openings are necessary, the head becomes unsymmetrical, and the greater the number of such openings, the more difficult it becomes to cast the head without shrinkage stresses, and the more liable is the head to heat failures resulting from uneven expansion and imperfect water cooling. In respect to both the number of such openings, and their distribution, the 4-cycle engine, and, although in a less degree, the overhead valve scavenging 2-cycle engine, are decidedly at a disadvantage, as compared with the port scavenging 2-cycle engine, in which the cylinder head requires only a single opening for the spray valve.

Fig. 43 shows a design of 2-cycle cylinder head in which there is only one opening for the combined spray and air starting valve. The head shown in Fig. 44 is of the 4-cycle

type. The openings for the different valve cages in the top of the head are grouped symmetrically; the spray valve in the center and the inlet and exhaust valves on each side. The opening for the starting valve is in front of the central opening, in the same plane, and at right angles to the plane through the other three valve openings.

A flat, smooth, bottom surface in a cylinder head results in a simple combustion chamber. Such a surface is a great

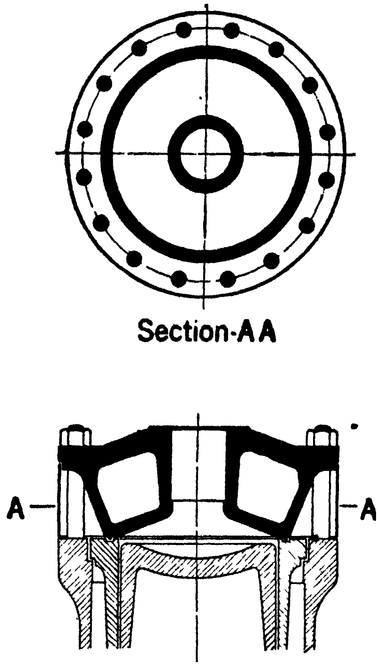


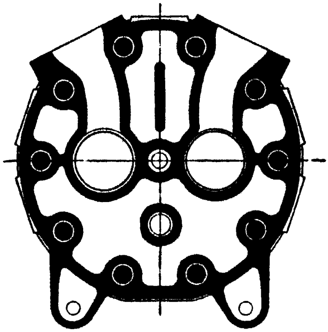
Fig. 43.—Type of 2-Cycle Diesel Engine Cylinder Head

aid in reducing the clearance space to the required minimum and effects favorably the combustion of the fuel. The advantages of a flat head are considered by some manufacturers of horizontal engines to outweigh those of vertically seated valves. Vertically seated valves are the most satisfactory and most builders of horizontal engines follow gas engine practice in the construction and operation of the valves and the cylinder head. The inlet valve is usually placed at the

top and the exhaust valve at the bottom with the spray valve placed horizontally in the center as shown in the Allis-Chalmers engine, Figs. 186 and 187.

Very few engines are now built having the valves seat directly on the cylinder head. The valves are placed in cages which are inserted into openings in the cylinder heads.

Pistons. Diesel engines up to approximately 22 inch diameter cylinders use the trunk type piston, which performs



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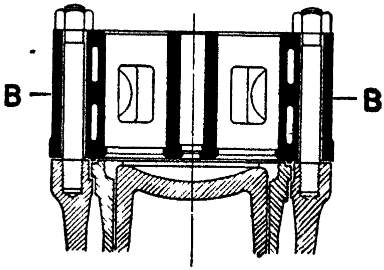


Fig. 44.—Type of 4-Cycle Diesel Engine Cylinder Head

the function of a cross-head and is, therefore, extra long. The material used is invariably close grained cast iron.

During manufacture the pistons are rough turned and as a safeguard against casting stresses are slowly and uniformly heated to slightly above their working temperature, at which temperature they remain for several hours before being slowly cooled again.

The final machining operations consist of a light cut, after which the piston is ground to size, proper allowance being

made for the different degrees of expansion of the various parts of the piston due to the heat transmitted from the combustion space. This allowance is shown diagrammatically in Fig. 45 in which the clearance is made several thousandths of an inch greater at the top, the theory being that the piston will assume cylindrical shape at running temperatures. Clearance is also increased at the wrist pin belt on some pistons to prevent piston seizures and undue wear on the cylinder.

A typical non-cooled trunk type piston is shown in Fig. 46. The piston is cast in one piece. The part of the piston wall at the top and down as far as the wrist pin bosses is quite thick and reinforced by strengthening ribs.

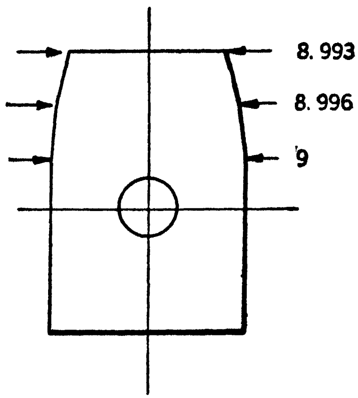


Fig. 45.—Heat Expansion Allowance of Piston

The wrist pin is held in place by the key, 1, and two set screws, 2. Oil for lubricating it is received through a hollow connecting rod.

The piston rings are six in number and of square section. The ring shown at the bottom of the piston is known as a "wiper" ring, its function being to scrape off the excess lubricating oil from the cylinder walls and cause the oil to drop back into the crank pit.

Non-cooled pistons are only suitable for small engines of less than twelve-inch bore. Even the smallest 2-cycle engines have some form of cooled pistons.

Cooled Pistons. Piston cooling may be classed in two systems, one in which the cooling is part of the lubrication

system, and the other in which it is separate, such as water cooling. A cooled piston of the former type is shown in Fig. 47. The upper portion of the piston head, 1, is cored out to form the cooling chamber. The oil, from the lubricating system, is forced up through the stationary tube, 4, and after passing through and cooling the wrist pin and its bearing, 3, makes its way up into the piston head. The oil discharges down through another stationary tube, 5. The piston, as it

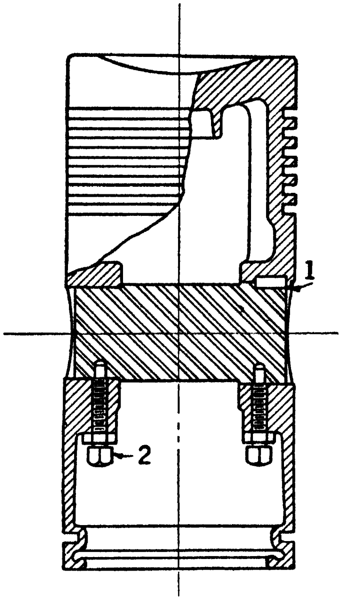


Fig. 46.—Uncooled Trunk Piston

reciprocates up and down, telescopes over the stationary tubes. The stuffing box, 6, is merely a loose fitting babbitt bushing which is not necessarily kept oil tight.

Piston Rings. Compression pressure mainly depends upon the fit of the piston rings, other conditions being normal. The clearance between the piston and cylinder walls varies of course with the size of engine. Piston rings should possess sufficient elasticity to press uniformly against the cylinder walls, and should be so located in the piston that no two joints come in line with each other. The rings should slide

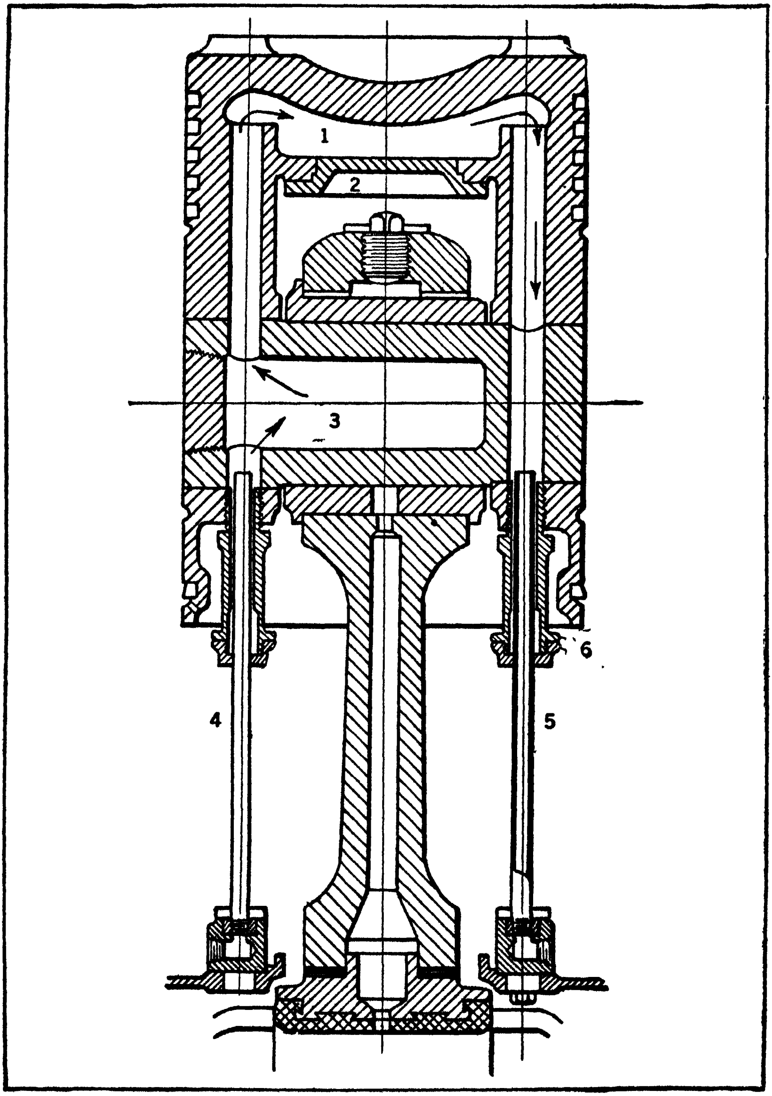


Fig. 47.—Oil Cooled Piston

freely in their grooves with no perceptible side movement. A dark discoloration along the outer circumference of both the piston and rings is a positive sign that hot gases of combustion have been leaking past them.

Piston rings should be examined to see that they have sufficient elasticity to function properly whenever a piston is removed. A weak ring may sometimes be improved by lightly peening it on its inner surface, the outer surface lying on a hard smooth foundation. This operation, which tends to expand the ring and cause it to press against the cylinder walls, must be carefully performed due to the fact that the rings, being of cast iron, are easily broken.

If the piston rings work around in their grooves so that their joints come in line with one another, piston leakage is

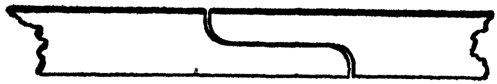


Fig. 48.—Lap Joint of Piston Ring

sure to result. To avoid this trouble some manufacturers fasten them in their proper place by small dowel pins secured in the piston. Shiny spots at the joints and diametrically opposite indicate that the ring does not properly fit the cylinder bore.

In manufacturing piston rings they are made slightly larger than the cylinder bore and then split. The joint most commonly used is shown in Fig. 48 and is known as the "lap" joint. After cutting the rings and forming the joint they are placed in a jig and their ends pressed together; the rings then being ground to a true circle. If the rings are split and not ground they will assume an elliptical shape when placed in the cylinder and will not fit properly. Properly fitted piston rings will show a bright surface all around their circumference.

Piston Seizures. Piston seizure is usually the result of too small a piston clearance. Lack of lubricating oil to the wrist pin will cause it to heat up and elongate and extend the piston in the vicinity of the ends of the wrist pin. This will

cause the piston to rub on the cylinder walls which is followed by a piston seizure.

Another cause of piston seizure is the sudden cooling of the cylinder walls which contract before the piston has cooled in proportion to the cylinder. This is often caused by suddenly speeding up the circulating pump or in running from warm river water into cold sea water without making proper adjustments to the circulating pump.

Cracked Pistons. In marine practice, cracked pistons are sometimes caused by the salting up of the piston cooling

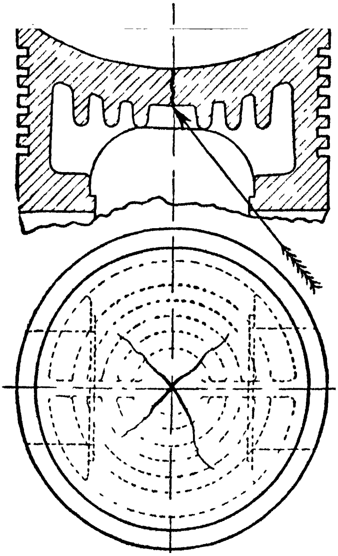


Fig. 49.—Common Form of Piston Fracture

system. Salt is deposited in sufficient quantities to clog up the pipes and passages in the piston head to a degree where the heat transfer is seriously interfered with, thus producing unequal heating and cracking of the piston. Most late marine installations now use fresh water for the cooling system, which is in turn cooled by salt water.

The center of the piston head, directly under the spray valve, is often raised to quite a red heat owing to direct contact of the piston head with the flame. After some time such over-

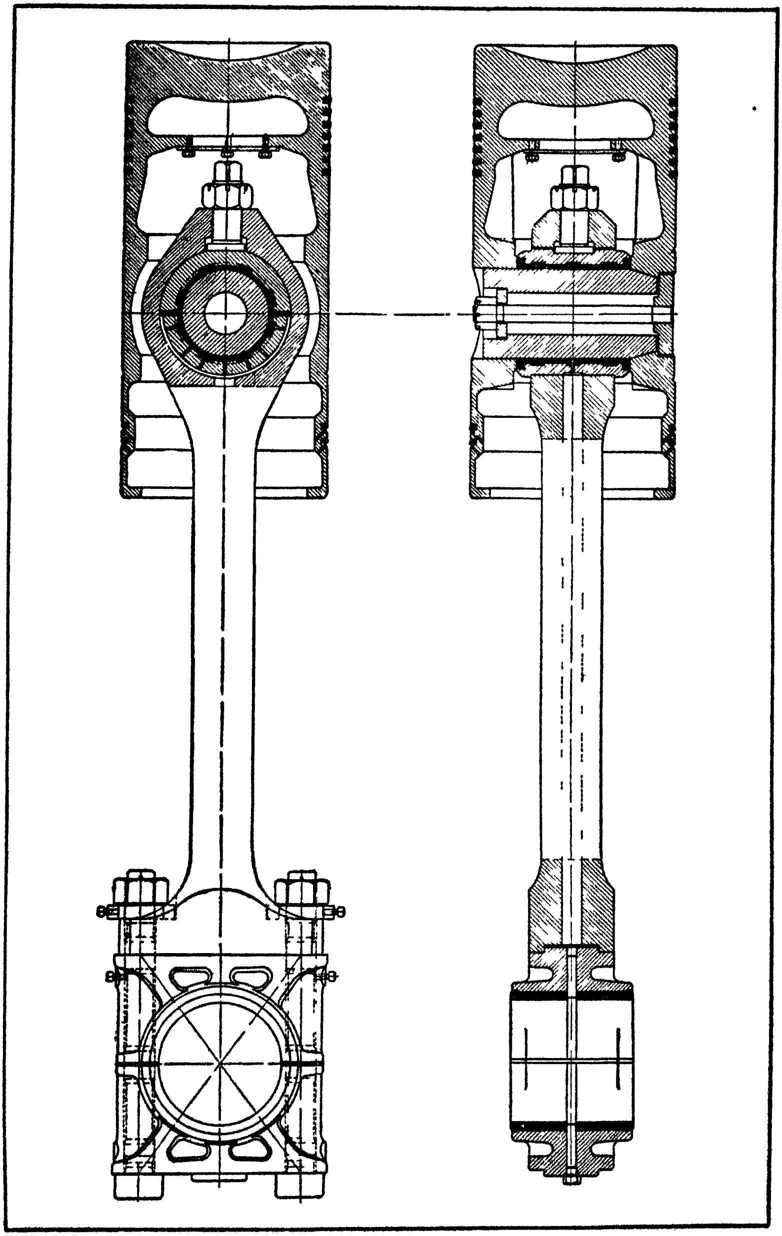


Fig. 50.—Typical Construction of Connecting Rod and Piston for 4-Cycle Engines

heated pistons become pitted and scaled in the heated area and frequently crack in a manner as illustrated in Fig. 49, the star cracks radiating from the center of the heated area.

Connecting Rods. Connecting rods are made of steel forgings of the same quality as those used in crankshafts. With engines using forced lubrication the connecting rods are usually bored hollow for the passage of lubricating oil and for lightness.

Fig. 50 shows a typical design of connecting rod and piston for 4-cycle engines (Worthington engine). The crank brass is made in two parts of cast steel, lined with babbitt metal. The two halves are firmly bolted to the foot of the connecting rod by two marine type crank pin bolts. Between the upper brass and the foot of the connecting rod shims are inserted which are used to adjust the cylinder compression. The piston pin end of the connecting rod is slotted out of the solid to

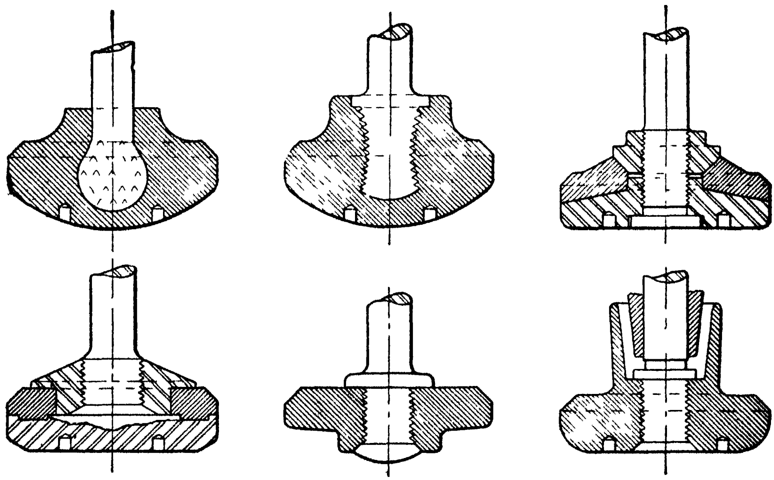


Fig. 51.—Various Forms of Exhaust Valves and Methods of Securing the Cast Iron Heads to Steel Valve Stems

receive an adjustable box which is lined with babbitt or else of solid bearing bronze. The halves of the bearing box are firmly clamped together by an adjustable screw in the upper end of the connecting rod. Wear of the piston pin bearing is taken up by scraping the abutting surfaces of the two halves of the bearing boxes.

The wrist pin bearing of 2-cycle engines is usually made in the form of a solid bronze bushing with no adjustment for wear. The connecting rods of crosshead type engines are of very similar design to those of marine steam engines, except that the wrist pin bearings are of more liberal dimensions to withstand the high pressures to which they are subjected.

Exhaust Valves. Exhaust-valve heads are usually of cast iron, which is less liable to become pitted than those made of steel. Typical methods of fastening the cast iron heads to the steel valve stems are shown in Fig. 51. The first two methods show the cast-iron head moulded around the steel stem. This is necessarily done at the foundry or place of

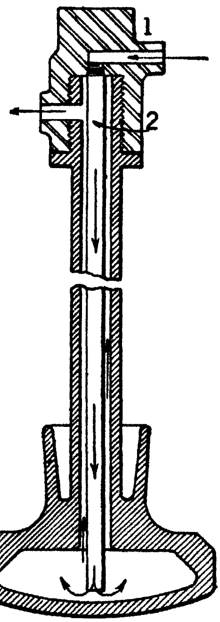


Fig. 52.—Water Cooled Exhaust Valve

manufacture. With the other methods shown the cast iron head is made separately and secured to the valve stem as depicted in the illustration. The last method shows an exhaust valve with an extension, or valve skirt, which overhangs the guide bushing of the valve cage and protects the steel valve

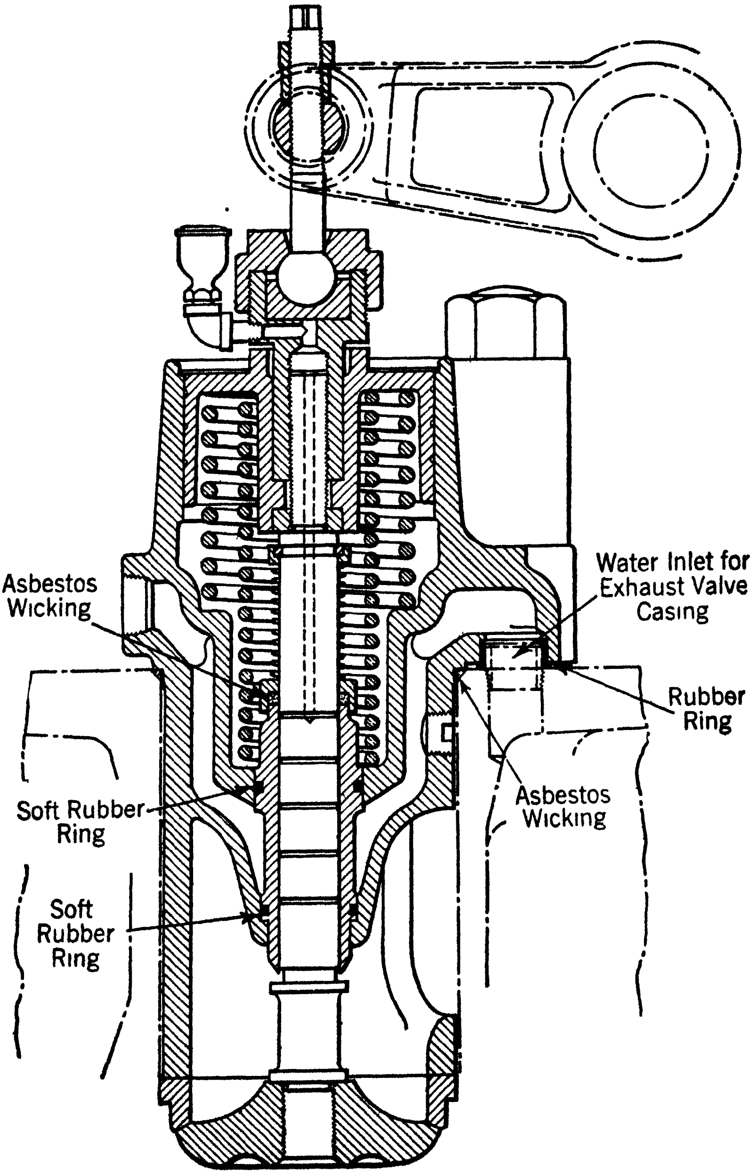


Fig. 53.—Section Through Worthington Exhaust Valve and Cage

stem from the hot exhaust gases when the valve is opened.

With larger engines some method of cooling the exhaust valve head is necessary, and this is accomplished much as shown in Fig. 52. A fitting, 1, is attached to the valve stem which is hollow, and circulating cooling water is led to and from the fitting by rubber hose. The cooling water passes through the inner tube, 2, and discharges at its bottom into the valve head. From here the water passes up through the space surrounding the inner tube and discharges out through the fitting as shown.

Valve Cages. In order that the valves and their seats may be readily overhauled and inspected they are usually contained in a separate casing or cage. Fig. 53 shows an exhaust valve cage and valve complete. The valve head is of cast iron and the stem is of forged steel threaded into the valve head with the end riveted over. The valve stem is of large diameter in order to promote rapid heat transfer. The valve seat is formed on a ring separate from the valve cage so that it can be readily renewed. The joints between the valve cage, valve seat and cylinder head are flat, so that in setting up for tightness no splitting stresses are set up in the head, as would be the case with tapered joints.

Attached to the upper end of the valve stem is a cylindrical guide of large diameter, which is housed in a bored extension of the valve casing and maintains the alignment of the valve and stem. The lower part of the valve stem is further guided by a long, removable, cast-iron bushing, at the upper end of which is a stuffing box and gland to prevent the lubricating oil being blown out by the exhaust pressure. This gland is held in place by a coil spring which bears against the lower end of the valve stem guide. This spring is compressed when the valve is opened, so that the maximum pressure against the packing is exerted only when the exhaust gases are passing through the valve. The valve cage has a water jacket which cools the valve stem and also the valve head, by heat transfer to the valve stem. The joints between the removable bushings and the valve cage body are made tight against water leakage by soft rubber rings held in circumferential grooves.

The bearing surface of the valve stem, where it is guided

by the removable bushing, is provided with circumferential oil grooves and lubricating oil is supplied through an axial hole from an oil cup on top of the stem.

Exhaust Valve Troubles. The principal troubles with exhaust valves are that they become pitted and worn and thus fail to hold the compression pressures. Valve cages that are not properly installed in the cylinder head often become broken by heat stresses. The clearances between the top of the cylinder head and the valve cage directly underneath the holding down nuts should be carefully measured with feelers when installing a valve cage and care taken to have the clearances equal when finally set up.

Trouble is also caused by the valve stem becoming gummed up with carbon and the valve stem sticking open. This can be overcome by allowing a good clearance between the stem and its guide bushing. The lubrication of these parts is difficult, and the expansion is considerable, causing a great deal of trouble from the stems sticking. With horizontal valves this clearance must necessarily be much less than with vertical. In either case the guide piston should be exactly centered as much depends on the alignment of the guide piston and valve stem.

Grinding Valves. To retain a perfect seat, the exhaust and inlet valves must be occasionally ground in their seats. This is accomplished by removing the cages from the cylinder head and the guide spring from the valve cage. The valve heads are usually provided with two holes for the insertion of a special fixture, or key, to turn the valve back and forth. Fine grinding compound is best used, finished by grinding with fuel oil. Valves and valve seats that are badly worn or pitted should have a light cut taken off, as attempting to remove deep pits and grooves by grinding tends to round the valve and its seat and only aggravates the trouble.

Inlet Valves. Inlet valves and their cages are usually of the same general construction as the exhaust valves except that the valve heads and stems are made of one piece of steel. Also, the valve cages and valve heads are non-cooled as these parts are amply cooled by the inlet air. Some builders make the inlet and exhaust valve cages interchangeable and in the

case of water cooled cages the inlet valves cage inlet and outlet cooling water passages are blanked off.

Air Suction Strainers. The intake valves of the working cylinders and first stage suction valves of the spray air compressor are usually fitted with some means of muffling the sound caused by the inrush of air through the intake passage ways and valves, as well as for excluding dust, dirt, waste and other foreign matter from entering the cylinders. Two common forms of suction strainers are shown in Fig. 54. They are built up of pipe or sheet metal and have a large number of slots about 1/32 inch wide, the aggregate area being from one and one-half to two times the area of the intake.

Spray Air Compressors. The engine air compressor, or spray air compressor, is used to furnish the compressed air

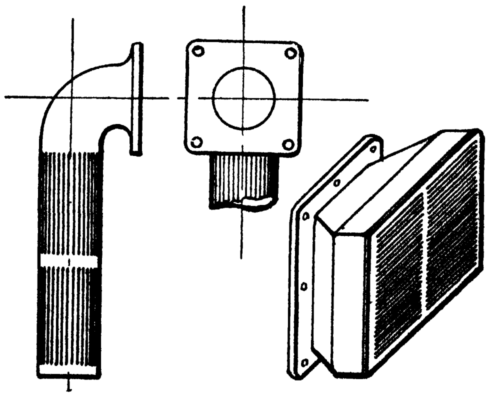


Fig. 54.—Air Suction Strainers

required to blow the fuel into the working cylinders. Besides furnishing the spray air at 600 to 1100 pounds pressure, it is also used to charge the air starting tanks as the compressor is made of somewhat larger capacity than that actually required to furnish the spray air.

The air compressor is usually driven direct from the main shaft of the engine, although a number of engines are built having the compressors driven by means of a beam from one of the cross-heads. On multicylinder engines with closed crank cases the compressor usually has the appearance of an addi-

tional cylinder, the compressor cylinders and air cooling pipes being surrounded by a jacket forming a water space such as shown in Fig. 183.

Some engines have two compressors to avoid the large size necessary with only a single compressor. Greater accessibility and reliability are thus obtained, as the work of compression is divided into a larger number of smaller impulses. Also, in case of a breakdown of one compressor, the other will furnish enough spray air to run the engine at slightly reduced power.

Compressors may be vertical or horizontal, and are usually built having either two or three stages. Three stage compressors are much to be preferred, as it permits compressing the air more gradually, and the lower compression ratios avoid excessive terminal temperatures, and also permits cooling the air more thoroughly in intercoolers before it is passed from one stage to the next higher.

Fig. 55 shows some of the most common arrangements of the spray air compressor. At A is shown a simple two-stage compressor, a design very common to small engines, but this arrangement and also the three-stage arrangement shown at C have a very serious drawback in that when the crank is on its top centre the pressure in each stage is at its maximum, and the total downward load on the working parts is equal to the load on each stage. The load is always downward, and is not reversed, thus rendering the lubrication of the various parts concerned much more difficult, as the continual downward pressure on the bearings ruins the oil film so necessary for efficient lubrication.

At D is shown another method of combining the three stages and at B a method of combining the stages of a two-stage compressor. With these two methods the maximum downward load is reduced and also an upward load brought in, which in turn has a lifting effect on the working parts and so helps their lubrication. With these two forms of compressors the second stage is formed by the annular space between the first stage portion of the piston. At D the third stage is formed by a separate small piston. The arrangement shown at E is sometimes used, the pistons being driven by cranks

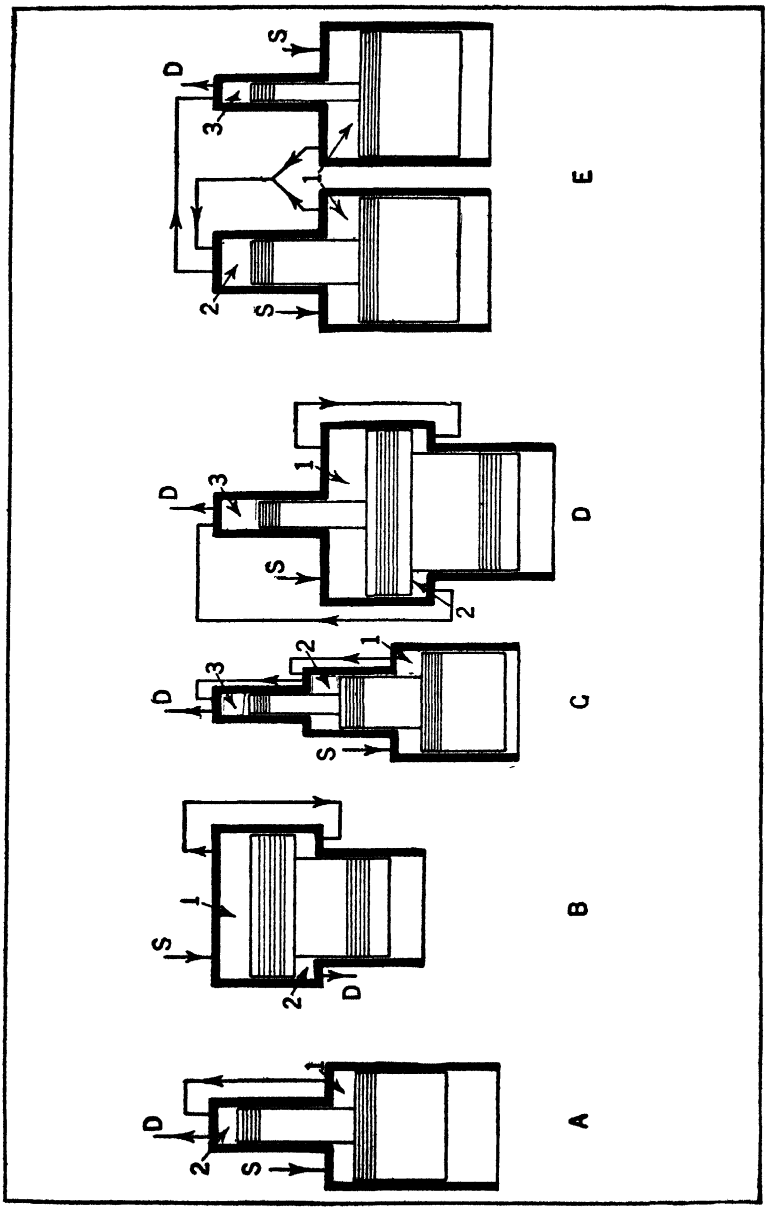


Fig. 55.—Various Arrangements of Air Compressors

180 degrees apart. Here the second stage piston is superimposed on one of the first stage pistons and the third stage superimposed on the other. Both first stage pistons discharge into the second stage as shown by the arrows.

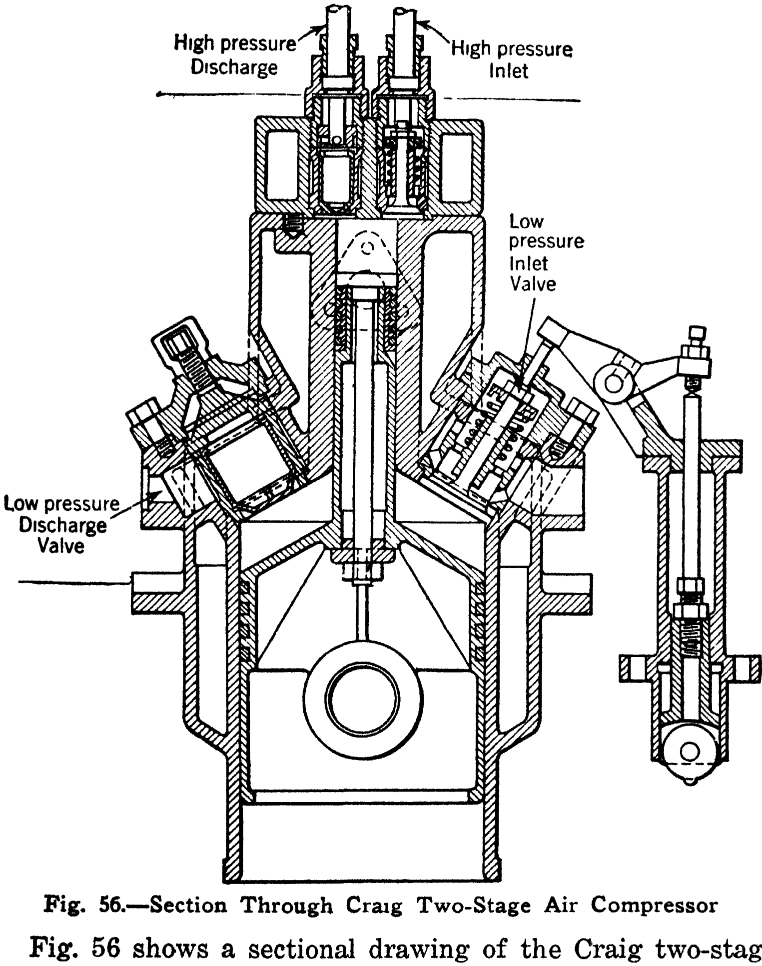


Fig. 56.—Section Through Craig Two-Stage Air Compressor

Fig. 56 shows a sectional drawing of the Craig two-stage compressor. The first and second stage pistons are cast in one piece and the second stage piston rings and spacer rings are held in place by a long bolt which passes through the second stage plunger and is secured by a nut on the inside of the first stage piston. The first stage piston carries rings in

ordinary grooves, much like that of a working cylinder piston, and the wrist pin and connecting rod are also carried in a similar manner.

Air compressor valves are of various types, the two forms shown in Fig. 56 being very common. The inlet valve of the first stage is mechanically operated and, like that of the second stage, is of the ordinary poppet type. The discharge from each stage cylinder is by a thimble valve which is held on its seat by a light spring. The valves, as common with most compressors, are housed in removable cages which facilitates their removal and replacement for overhaul and repair.

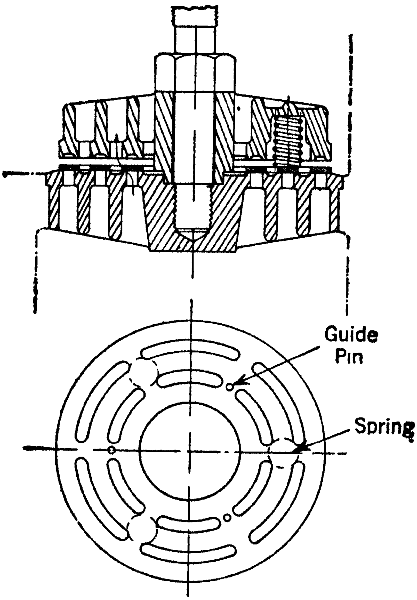


Fig. 57.—Air Compressor Disc or Plate Valve

Another very common form of compressor valve is shown in Fig. 57. This type of valve is known as a plate, or disc valve. It will be seen that the disc valve is provided with a number of circular ports, making it in effect a number of light rings joined at various points to provide the necessary stiff-

ness. The discs seat on very narrow faces of the valve housing, and as the area through the valve is equal to the total circumference of the ports, multiplied by the lift of the disc, it follows that very large areas are obtained for very small lifts of the valve. The cage above the disc valve is provided with a number of small springs of light load, while a number of dowel pins keep the valve properly located.

There are several types of air coolers. A common form is that shown in the sectional view of the Lombard engine, Fig. 188. This consists of coiled tubes of copper immersed in a cast iron casing through which the cooling water is circulated. Another form consists of a number of tubes held between two tube plates; the cooling water flowing through the tubes and the air passing around the outside of the tubes. Baffle plates are fitted to restrain the flow of the air and to obtain the full cooling effect from the tubes. Another form of cooler is very similar to this last, with the cooling medium on the outside of the tubes and the air passing through the tubes.

Where the air travels through the tubes, there is a danger on the higher stages that if a tube should carry away the high pressure air would leak out into the body or casing of the cooler and possibly cause serious damage. To prevent serious accidents of this nature the cooler body is usually fitted with a lead blow-out plate consisting of a circular plate of thin lead about two or three inches in diameter. In case of a serious leak the lead plate will be ruptured and the pressure released.

Moisture collects in the air as it passes from one stage to the other, and this moisture and also oil, is removed from each stage by separators below each cooler. The usual design consists of a heavy cylindrical steel body having a tube about three parts of the way down its center. The outlet for the air is at the top. The air enters through the long tube at high velocity, and on reaching the bottom of the tube has to turn quickly, and return up and out through the top discharge. In effecting the reversal of motion of the air any water or oil present in suspension is thrown to the bottom of the separator from which it may be drained periodically.

Foundations. Drawings and specifications for building foundations are always furnished by the manufacturers of Diesel engines to the purchasers. They consist of accurately dimensioned plates showing the location of the foundation bolts and a drawing of a templet to be used in locating these holes.

The size and weight of foundations vary with different types of engines, ranging from 8 cubic feet per brake horse power in the case of small high speed engines, to 20 cubic feet with large slow speed engines. With different conditions of soil and various designs of engines, each of which is subject to different degrees of shock from their unbalanced forces, a universal formula for engine foundations to apply to all engines would be impossible.

On firm ground foundations are usually made rectangular in shape and in "made", or yielding ground, it is advisable to expand the base of the foundation at an angle of about 30 degrees. Iron rods embedded vertically and horizontally,

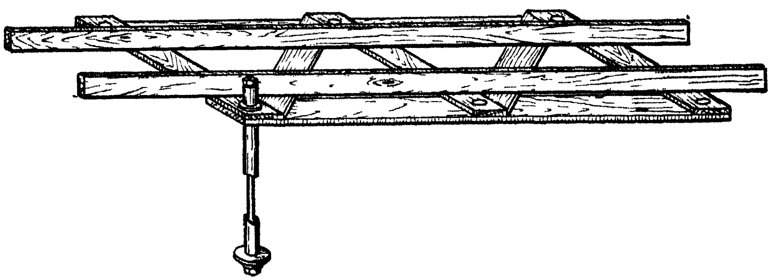


Fig. 58.—Wood Template for Suspending Foundation Bolts

adapted to the shape of the foundation and suitably tied together, form an inexpensive and satisfactory re-inforcement. Foundations should follow the general outline of the engine and include all machines and auxiliaries directly driven by the engine.

The concrete material varies with different kinds and grades of ingredients available. With clean sharp sand and washed gravel, a proportion of one part cement, three parts sand and four or five parts of gravel or broken stone is suitable.

The foundation pit should be deep enough to go well below the frost line of the locality. The bottom should be well rammed and large stones, a foot or so apart, placed for the bed. In made ground it is often necessary to resort to piling also.

A templet, showing the location of the foundation bolts, is made from the manufacturer's drawings or else laid off from the engine bedplate. The templet should be solidly built up of pine boards about one inch thick and eight or ten inches

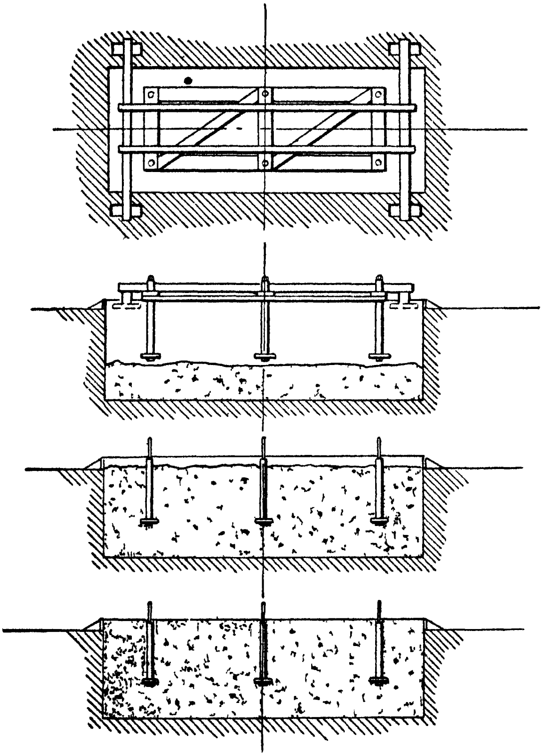


Fig. 59.—Steps in Pouring a Foundation

wide. Fig. 58 shows the construction of a templet for a small engine with the holes laid off for suspending the holding down bolts. These bolts are usually inserted inside a piece of iron pipe, the length of the pipe being such as to come just below

the surface of the finished foundation. This allows for the correction of any slight misalignment of the bolts after the concrete has set. The space inside of the pipes and around the bolts is filled with liquid cement after the bed plate has been placed in position.

After the pit for the foundation has been dug and partly filled with concrete the templet is placed in position and accurately leveled, somewhat as shown in Fig. 59. The center

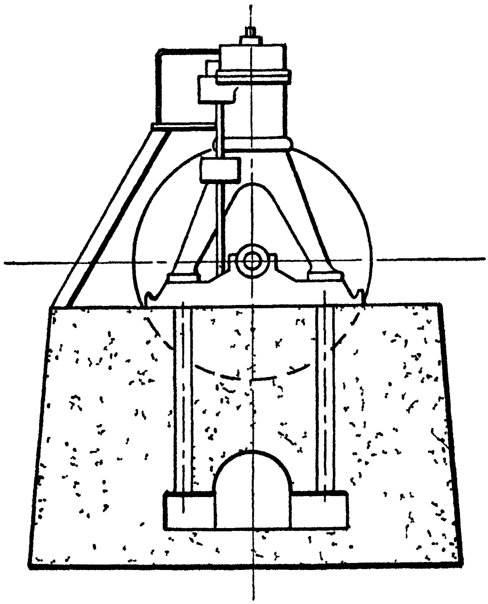


Fig. 60.—Foundation and Tunnel

line of the engine shaft, if it is required to be in alignment with another shaft, should be laid off with the aid of a Y-level, and a wire stretched to show the center line of the shaft. The location of the templet and the top of the foundation can then be measured from the wire.

The concrete should be poured into the pit in layers about six inches deep and each layer thoroughly rammed. If it is necessary to leave it uncompleted, as over night, it should be covered with wet sacks and before adding more concrete must be thoroughly raked over and slushed with cement.

When the foundation pit is nearly filled with concrete the templet is removed and the foundation brought to the correct height and leveled off. When the concrete has set, a period of a month or so, the bed plate is placed in position and the engine erected.

The foundations of large Diesel engines are usually built having a foundation tunnel as shown in Fig. 60. With this

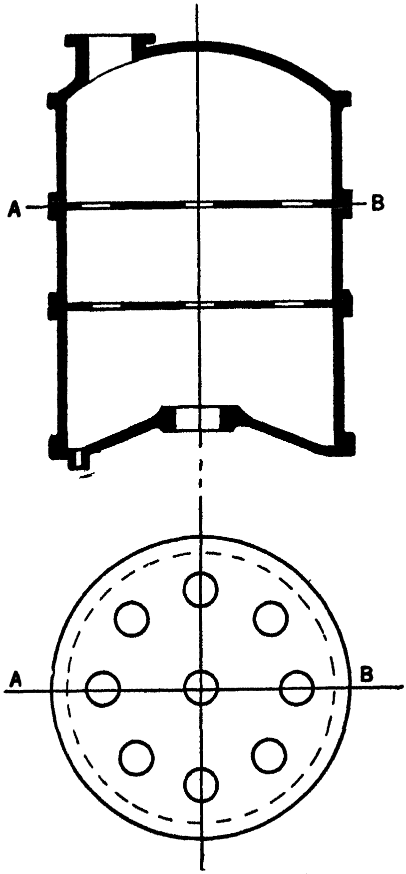


Fig. 61.—Exhaust Muffler

construction the foundation bolts can be placed in position after the bed plate is set in place.

Exhaust Mufflers. In order to reduce the noise of the exhaust from Diesel engines some form of muffler or silencer

is necessary. A simple form, built up of cast iron and having two baffle plates, is shown in Fig. 61. The baffle plates are of

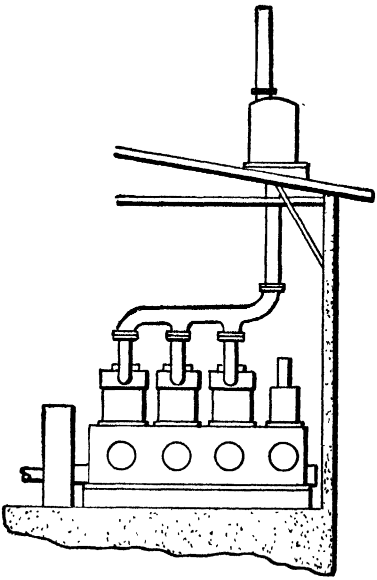


Fig. 62.—Muffler Erected in Place

sheet iron and have a number of holes in each to break up the exhaust. This type of muffler is usually erected on top of the

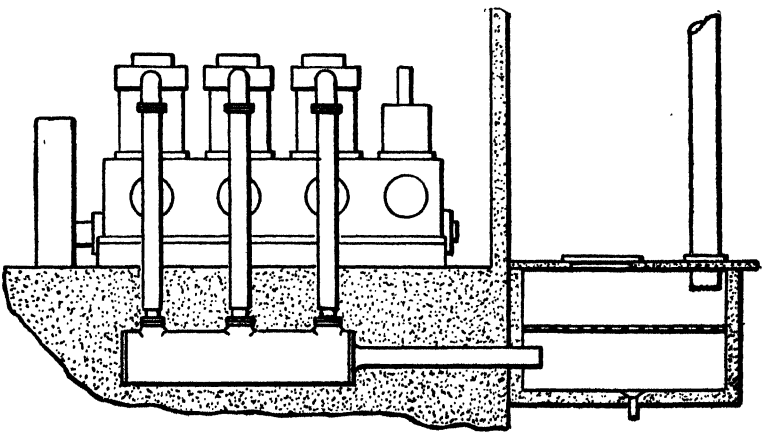


Fig. 63.—Exhaust Pit Form of Muffler

power house as shown in Fig. 62, the exhaust being conducted to the muffler through ordinary unjacketed iron pipe. Exhaust piping should be as short and have as few bends as possible. This is especially true with 2-cycle engines which must have a very low exhaust back-pressure.

A much more suitable arrangement is a form of exhaust pit or excavation as shown in Fig. 63. The exhaust gases from the engine first expand into a collector and then pass

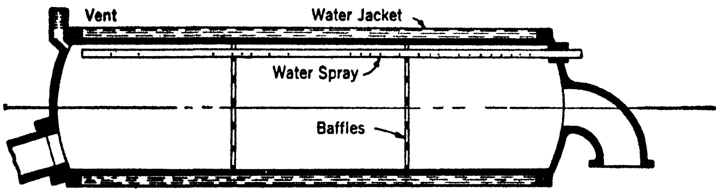


Fig. 64.—Type of Muffler Used on Submarines

on to the exhaust pit where they further expand. The pits have walls of concrete or brick. Baffle plates are sometimes installed but are usually unnecessary except where the noise of the exhaust must be completely silenced.

The noise from the exhaust of submarine engines must be silenced as much as possible. This is accomplished with a form of muffler as shown in Fig. 64. This is water jacketed and also has a water jet, or spray, in the interior of the exhaust passage. The exhaust gases, in passing through the curtain of water from the spray pipe, are cooled and condensed, and so effectively silenced.

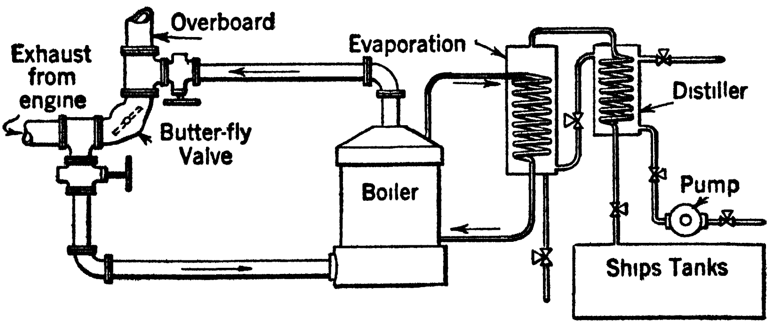


Fig. 65.—Exhaust Heated Evaporator

Exhaust Heated Evaporators. Ship and submarine installations are sometimes equipped with evaporators, which are heated by the hot exhaust gases from the main engines, for making fresh water from salt water. One system, as installed in American submarines, is shown diagrammatically in Fig. 65. When the engines are running the valves leading to and from the boiler are opened and part of the exhaust gases from the engine are made to pass through the boiler, the exact amount being regulated by a butterfly valve in the exhaust line between the two boiler stop valves as shown. The boiler is of simple tubular construction and contains fresh water which is used over and over again. The steam from this water, at about 30 pounds pressure, is led to the coils of the evaporator where it is condensed and returns again to the boiler as hot water. The salt water in the evaporator is heated by this steam and the steam and vapor from this salt water, at about 5 pounds pressure, is led to the distiller, where it is condensed into fresh water and run into the ship's tanks, the salt having been left behind in the evaporator. When the water in the evaporator reaches a certain density, a blow down valve is opened and all the water blown overboard. A fresh charge of water is then placed in the evaporator, being obtained from the pre-heated water of the distiller.

CHAPTER V

Spray Valves

Principles—Closed Types—McIntosh and Seymour Spray Valve—Nelseco Spray Valve—Worthington Spray Valve—Open Type Spray Valves —Koerting Spray Valve—Solid Injection Valves—Fuel Compression Chambers—Fuel Check Valves—Sighting Devices—Spray Valve Troubles and Adjustments—Adjusting to Viscosity of Fuels—Timing of Spray Valves—Spray Valve Cams.

The spray valve is the most important and most complicated valve on the Diesel engine and upon its correct functioning the proper and economical working of the engine depends. It performs two distinct functions. First, it has to admit the liquid fuel into the cylinder at the proper time, and therefore be an accurately timed valve, and second, it has to introduce the fuel into the engine cylinder in a completely atomized condition, and therefore must be an efficient atomizer.

One of the difficulties in its design is that only a small fraction of a second is available in which to inject the fuel gradually and in an atomized condition into the cylinder. The fuel must be converted in this short time from a viscous liquid into very minute particles, which have to be uniformly distributed in the injection air to form a proper mixture for sustained combustion. The fuel must also be delivered against the cylinder compression pressure of about 500 pounds per square inch.

The quantity of fuel oil delivered to the spray valve by the fuel measuring pump at each working cycle is very small, varying from only a few drops to about a thimble full.

Compressed air, at from 600 to 1200 pounds pressure per square inch is used for atomizing the fuel oil; incidentally it serves the purpose of injecting the fuel oil into the cylinder during the process of atomization. In the atomizing nozzle of the fuel valve part of the pressure of the injected air is changed into velocity, which reaches its maximum at the orifice of the nozzle. Owing to the expansion of the air at the orifice there is a decided drop in temperature at this

point, where the ignition of the fuel charge is to be initiated. This localized lowering of the temperature at the nozzle is accentuated during periods of low engine loads, when the fuel supply is diminished proportionally to the load, whereas the injection air supply remains constant; that is, the same as at full load, because the fuel valve needle stays open during 10 to 15 percent of the stroke of the piston, irrespective of the load. The relation of fuel supply to air supply is therefore altered with a change in load. It is consequently more difficult to start the ignition at partial than at normal loads. A fuel valve must be so constructed that unfailing fuel ignition at the proper instant will take place at every working impulse.

Spray valves are so designed as to introduce the fuel oil at a rate which allows the fuel to burn at constant pressure. If the fuel were injected all at once it would ignite in much the same manner as an explosion as in the surface ignition oil engine (Semi-Diesel Type).

Closed Type Spray Valves. A valve of this type is shown in Fig. 66. The fuel needle, or spray valve stem, 1, is guided in the atomizer sleeve, 2, and the cone, 3, and seats on the spray valve body, 4. The nozzle plate, 5, is held in place by means of a nut, 6. The nozzle plate is of casehardened steel having a small hole in its center. The nozzle plate projects into the combustion space of the cylinder.

The needle valve, 1, is lifted from its seat by the lever, 7. A hardened steel collar, 8, fitting into the lever, bears on the under side of the hardened nut, 9, which, together with a guide spring, 10, is screwed on to the needle valve. By adjusting the nut, 9 the correct clearance can be maintained by the spring, 11.

A light spring, 12, is interposed between the spray valve stuffing box, 13, and the atomizer sleeve, 2, to hold the latter in place and yet leave it sufficient freedom to expand. The high pressure air is prevented from escaping by means of the metallic packing, 14.

The atomizing elements consist of a slotted cone, 3, screwed on to the lower end of the atomizer sleeve, 2, and the atomizer plates and spacers, 15 and 16, respectively, which plates and spacers are clamped between the cone and the lower end of the

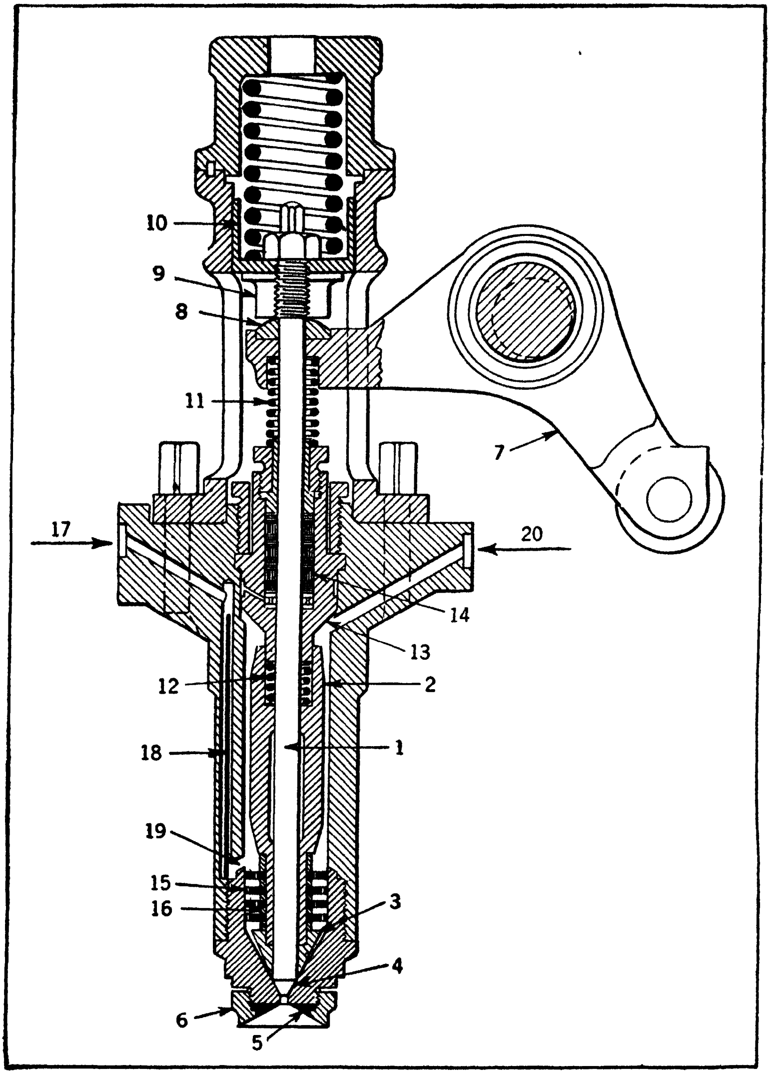


Fig. 66.—Closed Type of Spray Valve

atomizer sleeve. The atomizer plates are perforated as shown in Fig. 67 which also shows the shape of the cone. The rings of holes in the alternate plates are arranged in a staggered relation to each other.

In operation, the fuel pump delivers the fuel past the fuel check valves to the passage, 17. In the vertical passage a small wire, 18, is inserted to restrict the area of the opening, which causes the oil to be held in suspension, maintaining an equal supply to the atomizer at each stroke of the fuel pump. The fuel enters the annular channel, 19, overflowing from this on to the top of the uppermost atomizer plate. The quantity of fuel thus delivered for each working stroke is that discharged by the fuel measuring pump. The spray air enters the upper part of the valve body at 20, after first passing through an air check valve.

When the spray valve is ready to open, its entire body is filled with spray air, and the fuel oil lies on the upper atomizer plates. When the spray valve needle, 1, is opened by the

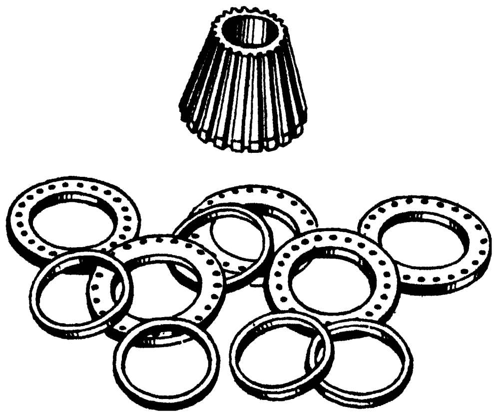


Fig. 67.—Atomizer Plates and Cone

lever, 7, the air blows through the annulus between the valve body and the atomizer sleeve, and forces the fuel oil through the holes in the atomizer plates and the slots in the atomizer cone and then through the orifice of the nozzle plate into the working cylinder. The passage of the air and oil, through the staggered holes in the atomizer plates and the slots in the cone, pulverizes the fuel, while the shape of the nozzle plate is such that the stream of air and atomized fuel is sprayed into the cylinder in umbrella form. The flow con-

tinues until the spray valve closes, fresh spray air following the mixture of air and fuel after the measured quantity of fuel has been delivered into the cylinder.

McIntosh and Seymour Spray Valve. The atomizer used on the McIntosh and Seymour engine is known as the "Hessel-

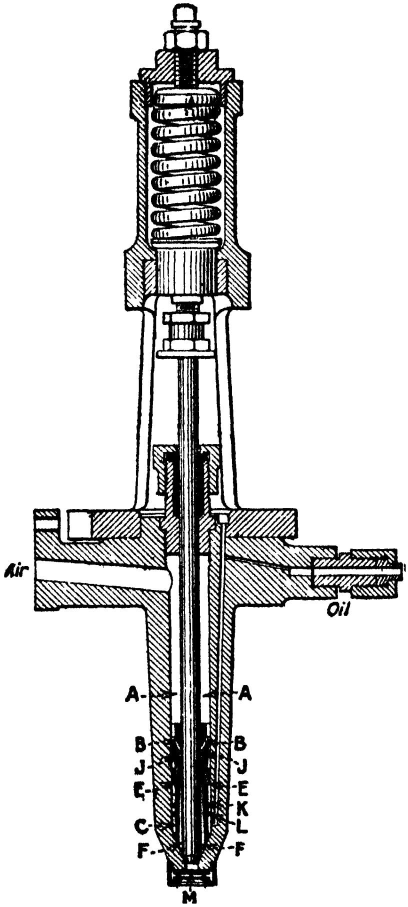


Fig. 68.—McIntosh and Seymour Spray Valve

man" type and is shown in Fig. 68. Its essential feature is that instead of pulverizing the oil by crowding it down through perforated plates it draws it by the injector principle into the current of in-going injection air, which atomizes and

absorbs it as fast as it is drawn up. The charge of oil is deposited in the chamber C, which it does not fill, even at an overload. Air is admitted to the chamber, A, and passes through the ports, B, while it has also access to chamber, C, through the space E-E. When the fuel valve opens, the air, rushing through the port, B-B, passes through the expanding passage, F-F, between the valve stem and the receding wall of the surrounding fitting, induces by injector action a difference of pressure which causes the fuel to enter the space, J-J, into which the fuel, having been elevated and broken up through the slotted plates, K and L, is drawn and picked up by the in-going air. The form of the fuel plate, M, has an important effect upon the efficiency of the atomizer.

It is claimed for this atomizer that fuel leaks into the cylinder are an impossibility, as the fuel cannot enter the cylinder, even with the needle valve open, until it is injected by the injection air.

The makers also claim that no alteration is necessary to any part of the fuel valve or atomizer, irrespective of the viscosity of the fuel used and fuels as low as 12 degrees Baumé are being burned in engines fitted with these atomizers, with equal success as with others using 26-30 degree Baumé fuel.

Nelseco Spray Valve. This type of spray valve belongs to what is known as the "straight-passage type," because the fuel does not pass through a perforated atomizer. In Fig. 69 the fuel enters at B, and is deposited by a distributor ring, C, in the annular passage, D, between the spray valve needle and valve body, and, when the needle valve opens the fuel is blown straight through, being effectively atomized by passing at high velocity through the chambers of changing diameter between the end of the spray valve needle and the atomizer plate, F.

Worthington Spray Valve. Fig. 70 shows the spray valve as used on the large 4-cycle Worthington engines. It is a long stem, steel disc valve working through a long guide and stuffing box on a conical seat. The valve body is in four main parts. The outer tube fits in the copper casing through the cylinder head and its lower end is provided with a square shoulder that makes a joint on a flat seat at the bottom of the opening through the head. This joint is made tight with a

McKim gasket. The inner tube, which forms a guide for the valve stem, is concentric with, and smaller in diameter than, the outer tube, so that there is an annular space between the two. The inner tube is accurately centered in the outer by the pulverizer rings at its lower end and a cover block into

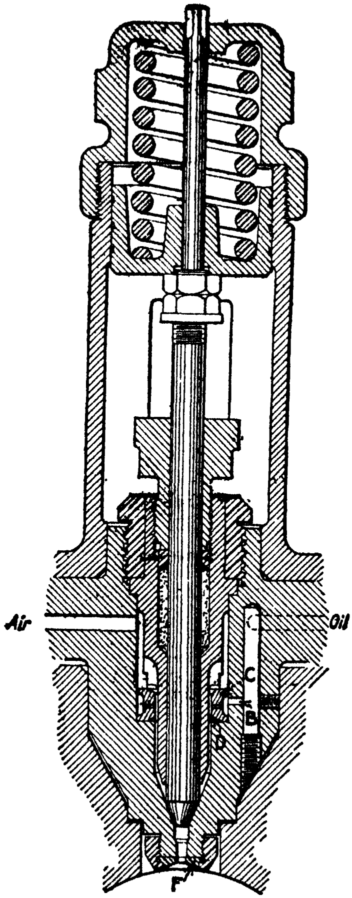


Fig. 69.—Nelseco Spray Valve

which its upper end is threaded. This cover block seats on top of the outer tube with a male and female ground joint. It contains the drilled passages through which fuel oil and spray air are admitted into the valve body.

Seated on the cover block, with a McKim gasket, is the valve cover, which is bored and threaded for a packing gland where the valve stem passes through. This valve cover is extended at the top to form a spring cage, bored at its upper end to receive a cylindrical guide of large diameter. This guide is threaded on the upper end of the valve stem. It guides the valve stem and its under side acts as a bearing for the valve spring.

The valve has a cone seat and the oil spray is discharged into the cylinder in the form of a diverging cone. The pulver-

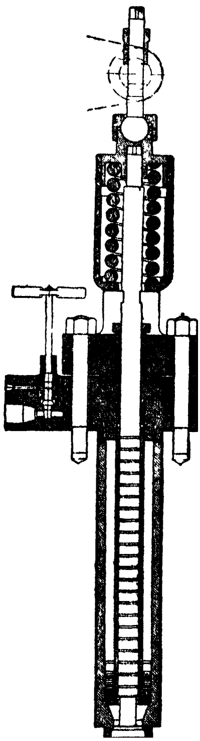


Fig. 70.—Worthington Spray Valve

izer through which the oil is forced before passing through the valve, consists of a series of thin discs, placed one on top of the other in the annular space between the inner and outer tubes, and separated by small shoulders around their inner

peripheries. Each disc contains a number of thin, radial slots and the discs are so placed that the slots are staggered, thus forming a tortuous path for the fuel.

Extending downward in the annular space between the inner and outer tubes, from the oil passage in the cover block to a point just above the upper disc, is a vertical pipe which deposits the fuel oil on the upper pulverizer disc. The annular space communicates with the air passage in the cover block, so that it is always filled with air under a pressure of from 800 to 1,000 pounds per square inch. The oil spreads out over

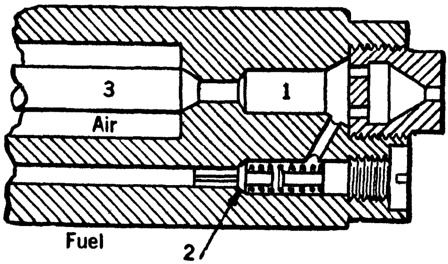


Fig. 71.—Open Type Spray Valve

the surfaces of the discs, and when the valve is pushed open by the valve lever the high pressure air rushes through the pulverizer discs and sweeps the oil into the cylinder in a finely divided form.

Open Type Spray Valves. These valves are so named because the fuel space is in constant communication with the cylinder. The valve is shown in principle in Fig. 71. Fuel is delivered into the fuel space, 1, past the check valve, 2, by means of the fuel pump. At the proper time the spray valve needle, 3, is opened by its valve gear and a blast of high pressure air blows into the cylinder and carries with it the fuel contained in the space, 1.

Koerting Spray Valve. The spray valve shown in Fig. 72 is of the open type. The fuel pump does not work against the high spray air pressure, but is subjected merely to the small pipe resistance and low lift necessary to deposit the measured quantity of fuel in a recess of the fuel valve chamber in front of the open nozzle. This is usually accomplished during the

suction stroke of the engine, when there is the least resistance, or air pressure, in the valve chamber. When the fuel is to be injected, the spray valve needle is opened and air is admitted to the fuel chamber. The air, in passing over the previously deposited fuel, impinges on it, picks it up and atomizes it into the combustion space of the cylinder.

Solid Injection Valves. In order to eliminate the cost, complication and danger of the high pressure air system required by the Diesel engine a great deal of development has been underway in the past few years towards injecting the liquid fuel without air. The advantages of such a simplifica-

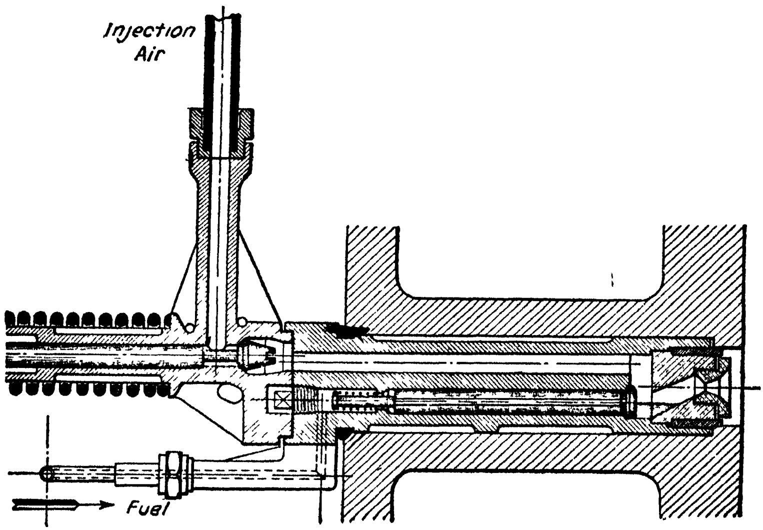


Fig. 72.—Koerting Spray Valve

tion of design are particularly noticeable for very small engines where the use of air injection would be prohibitive because of cost and the lack of care given these units.

The airless injection spray valve, when finally developed, will do much towards the perfection of the Diesel engine. The power lost in the spray air compressor amounts to as much as 20 per cent. in some engines and in most designs is also the source of considerable trouble. Most of this power could be

saved by the perfection of the solid injection spray valve. Also, ignition is usually arranged to take place at lower compression pressures.

In general, two systems are used in solid, or airless injection. In one, fuel is pumped to the spray valve at a pressure of 4,000 to 5,000 pounds per square inch, and is admitted when the valve is opened by suitable cams. The quantity of fuel admitted to the cylinder varies with the time that the fuel valve is open, and thus with both the cam profile and the engine speed. The second system uses a separate fuel pump for each cylinder, which delivers the desired quantity of fuel at whatever pressure is necessary.

Fig. 73 shows a valve of the second class. The spiral passages, 1, are used to secure a turbulent, quickly spraying stream of fuel and the spring loaded check valve, 2, prevents fuel remaining in the fuel line from "dribbling" in after the desired quantity has been forced through.

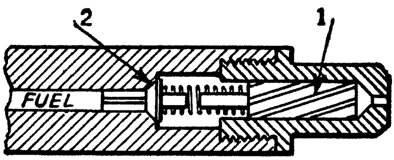


Fig. 73.—Solid Injection Spray Valve

Fuel Compression Chambers. Most engines using "solid" injection, use some form of compression chamber in the form of a tube, and having collapsible walls which expand under the high pressure of the oil being forced into the tube. When the fuel valve opens the tube collapses, forcing the fuel under high pressure into the cylinder. The pressure tube is usually elliptical and is forced into a cylindrical shape under the pressure of the fuel of about 3,000 or more pounds per square inch. Distance pieces prevent the tube from expanding or collapsing to too great an extent.

Fuel Check Valves. In order to prevent leakage of high pressure air from the spray valve to the fuel lines leading from the fuel measuring pump, a fuel check valve, such as shown in Fig. 74 is placed in the fuel delivery passage, close

to the spray valve. The check valve is kept seated by a spring with the spray air pressure acting on top of the check valve. The fuel pump delivering fuel to the spray valve must force this check valve open against any pressure that may be on it. Also, in the same fuel check body is a by-pass valve, which, when open, allows the fuel pump to discharge oil to an over-

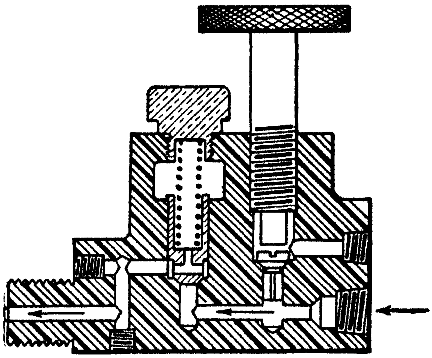


Fig. 74.—Fuel Check Valve

flow fuel line leading back to the fuel pump or to a tank. On closing the by-pass valve the passage is closed and the fuel is forced past the check valve into the spray valve.

The overflow pipes usually terminate in a "tell-tale" such as shown in Fig. 75. By opening the fuel check by-pass the delivery of fuel by the fuel measuring pump can be noted and air can also be worked out of the fuel lines by opening this by-pass.

Trouble with cylinders mis-firing can be quickly traced to the fuel measuring pump by opening the by-pass on each spray valve in turn and noting the amount of fuel pumped through the telltale.

The most frequent failure to obtain ignition in the working cylinders is the presence of air in the fuel lines to the spray valves. Since the small quantity of fuel required for each impulse stroke is forced into the end of a small pipe distant from the spray valve it is essential that the pipe throughout its length be filled with incompressible fluid if any is to be forced out at the other end against the pressure of the spray

air. The presence of air in the pipe results in compression and expansion of air in the pipe instead of ejection of fuel from the pipe.

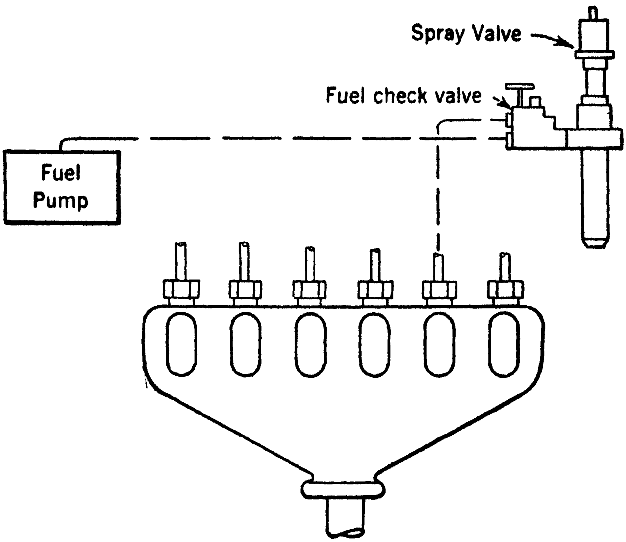


Fig. 75.—Fuel Oil Sighting Device

Air in the fuel lines is always due to leaky fuel check valves. These have a most difficult function. They must admit the fuel oil to the pressure chamber of the spray valve and must seat easily and tightly to prevent air from leaking back into the fuel lines. Each little increment of air coming through one of these valves increases the compressibility of the fuel in the fuel line and decreases the amount of fuel pumped into the spray valve at each stroke until finally that particular cylinder ceases to fire. As additional air leaks in it finally works back to the fuel measuring pump chamber and gets into the other fuel lines and stops the engine.

Spray Valve Troubles. The spray valve requires careful attention. The valve stems must work freely in the stuffing boxes at all times because if they happen to stick open, explosions are likely to result in the cylinder. When this happens, the fuel is sprayed into the cylinder as fast as delivered by the fuel measuring pump to the spray valve. This causes

pre-ignition and high temperatures and pressures in the cylinder which cause dangerous loads to fall upon various parts of the engine.

The spray valve should be carefully ground in, using very fine emery or ground glass. Care should be taken to get a bright seat with an even bearing all around. After grinding it should be thoroughly washed with fuel oil or kerosene in order to remove all traces of the grinding material. Unnecessary grinding in of the spray valve should be avoided and it is only necessary when it becomes worn and leaks. To test the spray valve for leaks the engine is jacked over to such a position that all spray valves are closed, then, by putting an air pressure on the spray air header and opening the indicator cocks the valves that are leaking can be detected by the hissing of escaping air.

To secure a tight valve stem as well as good combustion, it is necessary that the valve stem be exactly centered in its seat. To verify this, blue (Prussian blue) the stem of the valve and move it up and down a few times without turning. The blue should then show all the way around the valve guide. Before doing this the guide must be carefully wiped off.

Spray valve packing also gives considerable trouble. Usually a graphite-metallic packing is used. The packing must be tight enough to prevent the high pressure air from escaping, and yet if set up on too much it will cause the valve to stick open. When packing a valve, care should be taken to completely fill the stuffing box, as otherwise the valve stem is liable not to be held straight and will leak. If the stem has too much play in its guide more packing must be used. Small air leaks around the valve stem immediately after packing are harmless and generally cease after a little use.

Operating routine should require that the seating of the spray valves be tested at frequent intervals by the engineer feeling the exposed end of the spray valve stems and spray air piping as well. If the valve is seating properly the touch will indicate metal-to-metal contact.

It is also a good plan to turn the valve stems frequently on their seats while operating in order to keep them working properly and to even up the wear on their seats.

There are certain troubles very common to spray valves. The small holes and grooves become stopped up, especially if the fuel oil is not properly strained. Gumming up of spray valves is often caused by lubricating oil in the spray air compressor which is broken up into tar-like deposits by the combined heat and pressure of the air. Another cause of trouble is due from carbon deposits, caused mostly by improper combustion of the fuel oil, which may be due to poor fuel or low spray-air pressure.

Spray nozzles sometimes fall off or crack. This will cause poor combustion and a black smoky exhaust owing to the improper atomization of the fuel.

Adjusting Spray Valves to the Viscosity of Fuels. In changing from one grade of fuel to another it is sometimes necessary to vary the number of atomizer plates in order to obtain proper combustion. When very light fuel is used, more atomizer plates are usually required than with heavier fuel. The number of plates must be determined by trial, and decided on the basis of the number which gives the clearest exhaust.

Timing of Spray Valves. Spray valves usually open from 3 to 12 degrees before top center on the compression stroke, the exact time varying with different makes of engines.

In consequence of minute variations in clearances and cam forms, absolute uniform valve settings cannot always be obtained; and the differences are made very apparent by the magnified readings on the rim of the fly-wheel.

When timing an engine, the spray valves should be timed for opening first. All other valves are subservient to this if attached to the same cam shaft. All valve settings should be performed with the engine in a cold condition. The crankshaft should be barred in the direction of rotation of the engine. If a mark is over run, the crankshaft must be barred back well beyond the mark, and then, in the proper direction, again to the mark. All valves should be set for their opening points only.

To determine the time of opening of a spray valve the spray air header is charged with air and the engine barred over slowly. By listening at the indicator cock the time of

opening can be detected by the escape of air when the valve starts to lift, and the degree of opening can then be noted from the degree marks on the flywheel rim.

Slight differences in timing can sometimes be made by increasing or decreasing the cam clearance. This is done by giving more or less clearance to the tappet bolts on the rocker arm. This method of varying the timing can not be carried out if the timing is out of place more than two or three degrees as it will effect the lift of the valve.

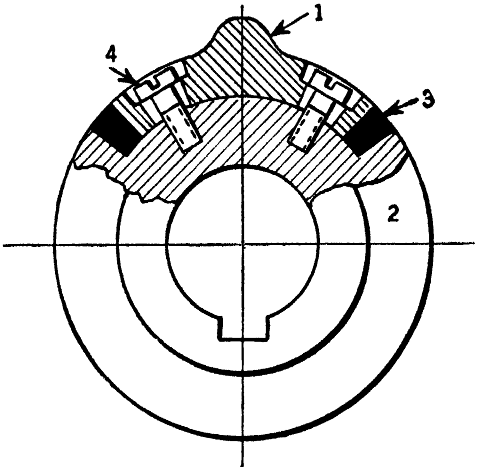


Fig. 76.—Adjustable Nose of Spray Valve Cam

Most spray valve cams have adjustable nose inserts by which the timing can be varied a number of degrees. Such a cam is shown in Fig. 76. The adjustable nose insert is shown at 1. The cam body, 2, is secured to the cam shaft by means of keys or it is made an integral part of the shaft. To adjust the position of the cam nose, shims of thin metal are placed between the wedges, 3, and the cam nose insert. The screws, 4, hold the insert firmly to the cam body.

CHAPTER VI

Fuel Pumps—Governors—Fuel Systems

The Function of the Fuel Pump—Variations of Fuel Delivery—Constant Stroke Plunger Types—Variable Stroke Plunger Types—Number of Pumps Required—Nelseco Fuel Pump—Busch-Sulzer Fuel Pumps—Worthington Fuel Pumps—McIntosh and Seymour Fuel Pumps—Other Makes of Fuel Pumps—Types of Fuel Pumps that Supply Fuel to Open Type Spray Valves—Troubles of Fuel Pumps—Regulating Quantity of Fuel Pumped—Care and Adjustment—Timing and Setting of Valves—Governors—Sensitive Governors—Trip Governors—Marine Type—Stationary Types—Lubrication of Governors—Fuel Oil Systems—Fuel Oil Compensating Systems of Submarines—Strainers, Filters, etc.—Gravity Tanks.

The function of the fuel measuring pump is to take fuel oil from the suction, or gravity tank, and deliver it in exact quantities to the spray valve against the spray air pressure. These pumps are always of the plunger type because they must pump against pressures of from 600 to 1,000 pounds pressure per square inch. They are quite a complicated piece of mechanism, containing some very accurate adjustments, upon which the proper and satisfactory operation of the engine depends.

Each working cylinder of a Diesel engine must receive exactly the same proportion of fuel as the other cylinders and some very close adjustments have to be made to the fuel pump to accomplish this. A fuel delivery of 2 c.c. corresponds to a full load injection for a 50 H.P. cylinder having about 80 injections per minute. A load variation of 1 per cent. amounts to a change of about .02 c.c. in the fuel delivery of the fuel pump. It will thus be seen how accurately the fuel pump must deliver minute quantities of fuel.

This small variation in fuel delivery was accomplished in the older types of engines by regulating the stroke of the pump plungers. The method was commonly to have the eccentric driving the pump act through a movable link or lever by shifting the fulcrum or position of which the plunger stroke

may be varied, thus pumping a varying quantity of fuel at each stroke. The main objection to this method was that it required so much power to shift the plunger stroke when the pump was working against its high pressure that it was very difficult to govern with a sensitive governor. Also, when running at light power the plunger stroke was so short that trouble was always caused by the valves failing to operate properly.

The system in almost universal use is to have a constant stroke plunger, the quantity of fuel delivered being regulated by the time of closing of the fuel pump suction valves. These pumps furnish a quantity of fuel exceeding the maximum amount required by the engine when working at full power. The excess fuel is forced back into its suction chamber, the suction valve being held open, the point of its closing commencing the delivery of fuel to the spray valve. The pump plunger thus delivers only the exact quantity of fuel required by the engine load. The suction valves are under the control of the governor and in case of excessive speeds the governor acts upon the suction valves, holding them open and preventing any delivery of fuel to the cylinders.

Fuel pumps are usually built from a heavy steel block forging drilled out for the pump barrels and valve openings. Some pumps have bronze or cast iron castings for the pump body. The plungers are of steel, hardened and ground and usually lapped into the pump barrels. The valves, one inlet and one or two discharge, are of steel or composition, cone seated and of the automatic type with the exception of the suction valve which is mechanically operated and controlled for regulating the quantity of fuel pumped in order to govern the speed and power of the engine.

Number of Pumps Required. When only one fuel pump is supplied with an engine, the fuel has to be divided equally to each of the working cylinders. This is accomplished by means of small steel diaphragms, the size of the holes being determined experimentally. This method has not proven satisfactory and most builders use a separate pump for each cylinder, the plungers being driven in groups of one, two, three or four from one eccentric. The best practice is considered that of

driving the plungers in pairs from one eccentric as the fuel can then be forced into the spray valves at a certain period before the opening of the spray valve. This insures that the action of the working cylinders so far as the supply of fuel is concerned will be the same.

Types of Fuel Pumps. Fuel pumps are divided into two distinct types; the high pressure type which supplies fuel to the closed type of spray valve and the low pressure pump which supplies fuel to the open type spray valve. The low pressure pump operates only against the friction of the piping to the spray valve while the high pressure pump has to operate against the high spray air pressure, the friction of the piping and the load of the check valve springs. The high pressure type of fuel pumps will be first taken up.

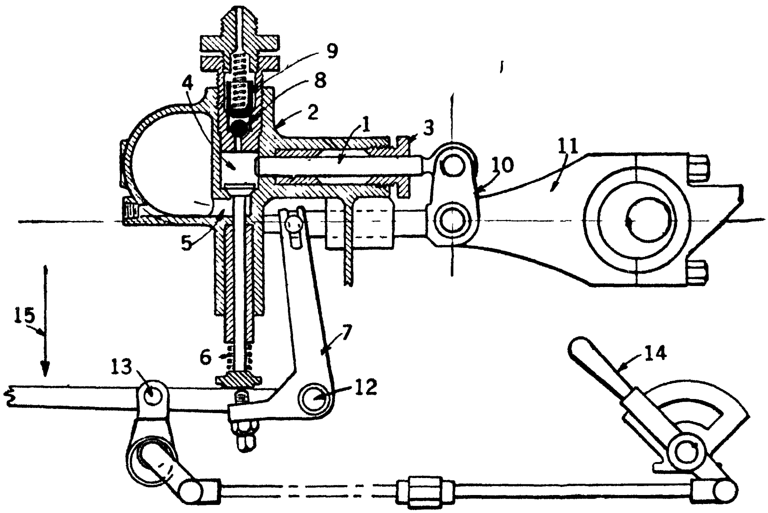


Fig. 77.—Nelseco Fuel Pump

Nelseco Fuel Pump. Fig. 77 shows the general mechanism of a fuel pump of a Nelseco engine. There are six plungers in a row in this pump. The plungers, 1, are driven in pairs from three cross-heads and eccentrics by one of the cam shafts. The plungers operate in holes, or barrels, drilled in the heavy steel casting forming the pump body 2. The pressure of the pump is held in by the metallic packing and the packing nut 3. At

the lower part of each fuel pump chamber is the suction valve which opens from the fuel oil suction chamber 5, containing a supply of fuel received from the gravity tanks. The suction valve is held shut by a light spring 6, and is opened on the out stroke of the plunger either by the suction of the plunger or by the bell crank 7, which is also operated by the cross-head which drives the pump plunger. At the top of the pump chamber is the discharge valve. This consists of a small ball check 8, and a regular poppet discharge valve with a spring above it. From this point a small pipe carries the fuel to the spray valve of that particular pump.

The operation is as follows: As the plunger is carried outwards by the cross-head 10, and eccentric 11, it will take in fuel through the suction valve. The amount of fuel taken in on the suction stroke is about twice as much as the spray valve will receive when the engine is running at full power. As the plunger starts back on its inward, or discharge stroke, the suction valve is held open by the bell crank during the commencement of the stroke and therefore the fuel will be, at first, merely forced back through the suction valve into the suction chamber as there is considerable pressure holding the discharge valve closed. At about mid-stroke (if the engine is running at full power) the bell crank will finally swing back far enough to allow the suction valve to seat, whereupon, during the remainder of the stroke the fuel will be forced past the discharge valves to the spray valve.

It will be seen that the end of the bell crank which strikes the suction valve tappet moves regularly up and down as the crosshead moves to and fro at each stroke. The bell crank has a pivot or fulcrum at the point 12 which is upon the large control arm, which also has a fulcrum at 13. By moving the hand lever 14, the fulcrum of the main control arm (at 13) carrying the bell crank lever can be raised. This causes a change in the point where the bell crank lever allows the suction valve to seat.

If the engine should race suddenly beyond the speed at which the governor is set to act, the governor will bear down on the control arm at a point beyond 15. The control lever, pivoting at 13, will raise the bell crank lever 7 upwards, hold-

ing open all the suction valves and cutting off the supply of fuel to all the working cylinders until the speed again falls to normal.

Busch-Sulzer "Type B" Fuel Pump. An extremely simple design of fuel pump and a type adopted by many English and European builders is shown in Fig. 78. The pump plunger 1, is driven by means of the eccentric 2, and guide piston 3, from the vertical cam driving shaft. The suction valve 4 is mechanically operated by the crank 5 which is free to swing around a pin supported by one end of a fulcrumed arm, the other end of which is connected with the governor. The push rod 6,

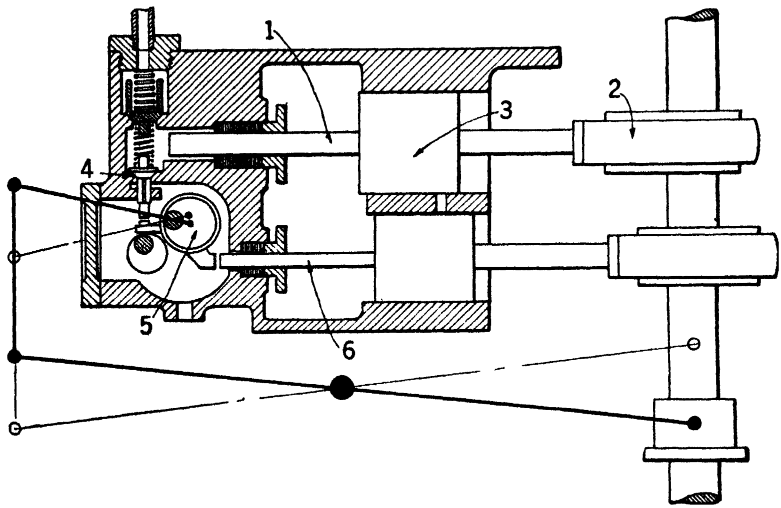


Fig. 78.—Busch-Sulzer Type "B" Fuel Pump

with constant stroke opens the suction valve by lifting crank 5 around its point of support; the magnitude of the lift is controlled by the extent that 5 is lifted under the influence of the governor. The lower the load the greater is the lift of 5, and, correspondingly, of the suction valve, and a greater quantity of the fuel by-passed. Although the stroke of the push rod remains always constant, the surface of crank 5 in contact with the end of the push rod slides past it up or down when being lifted or lowered by the governor.

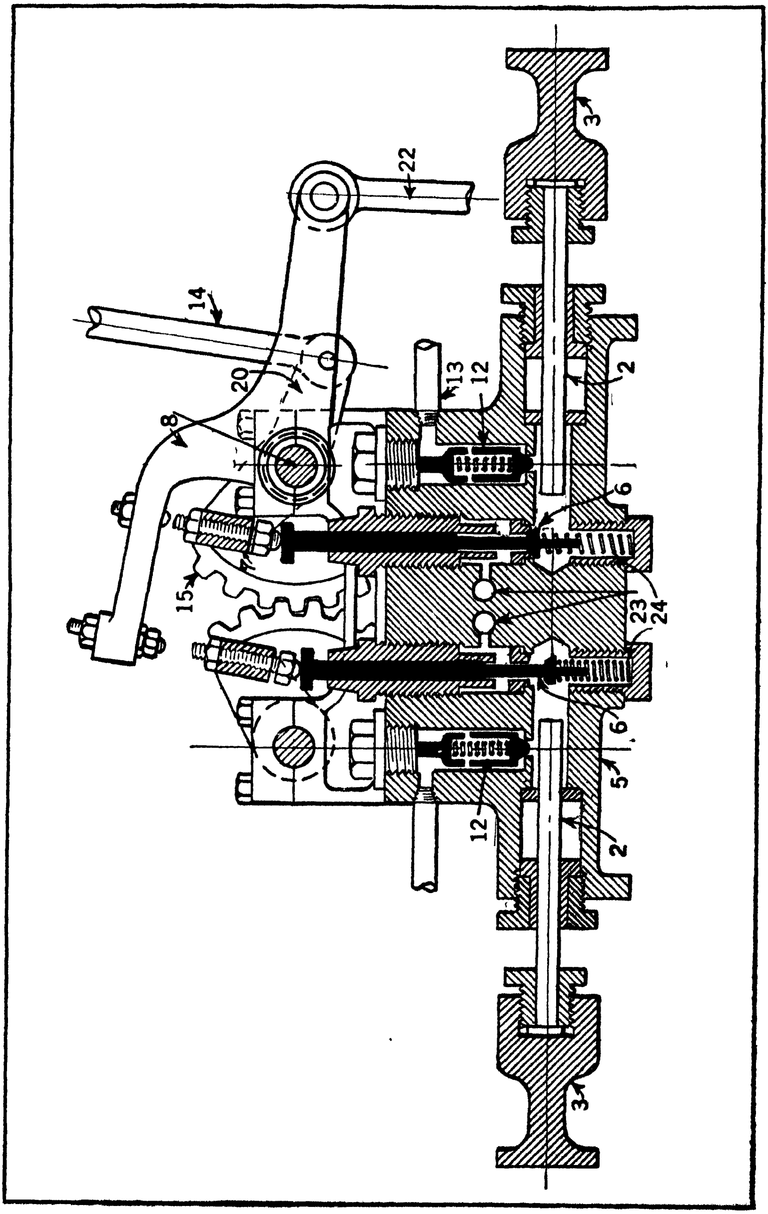


Fig. 79.—Longitudinal Section Through Busch-Sulzer Marine Type Fuel Pump

Busch-Sulzer Marine Type Fuel Pump. The construction and operation of the Busch-Sulzer light-weight marine type fuel pump is shown in the longitudinal section through the pump, Fig. 79, the perspective front view of the pump, Fig. 80, and the operation of the fuel pump suction valves, Fig. 81.

The fuel pump is operated by an eccentric, 1, mounted on the vertical cam drive shaft, at the after end of the engine, the eccentric making one revolution for each revolution of the crank shaft. The pump is provided with six plungers, 2, each plunger serving an individual cylinder. The plungers are arranged in sets of three, one set being on each side of the pump.

The plungers are of steel, hardened and ground, and are mounted on the driving beams or yokes, 3, which are connected by the side rods, 4. The plungers, yokes and rods form a rigid combination, which receives its reciprocating motion from the eccentric, 1.

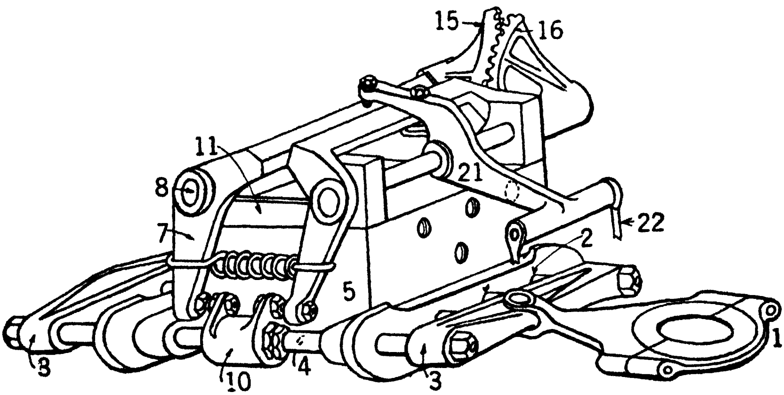


Fig. 80.—Perspective View of Busch-Sulzer Marine Type Fuel Pump

The pump body, 5, is a forged steel block, drilled to provide the necessary passage for the in and out-flow of the fuel oil, and to receive the plungers and valves. A separate passage delivers the fuel to each individual suction valve; and a separate passage discharges it from each individual discharge valve.

During the outstroke of a plunger, fuel is drawn into the plunger barrel, through the suction passage, 23, and the suction valve, 6, the suction valve being held open by the rocking lever, 7, pivoted eccentrically upon the shaft, 8. The rocking

lever, 7, receives its oscillating motion from the block, 10, which is rigidly mounted upon one of the side rods, and provided with adjustable tappet screws, which push against hardened screw heads on the lower ends of the rocking levers, 7; the parts being held in contact by the spring, 11.

During the instroke of the plunger, the fuel is discharged from the plunger barrel, back through the suction valve, until the rocking lever, 7, has moved away sufficiently far to permit the spring, 24, to close the suction valve; the remainder of the fuel is then forced out of the plunger barrel, past the discharge valve, 12, through the discharge pipe, 13, to the fuel spray valve on the working cylinder.

The quantity of fuel delivered to the spray valves, therefore, depends upon the period during which the suction valves are held open by the rocking levers, 7, while the plungers are making their discharge strokes. This period is adjusted to suit the desired engine speed by the fuel control lever at the operating station. The movement of the fuel control lever is transmitted, through the control rod, 14, to the lever, 20, which is keyed to the shaft, 14. The teeth of the gear segment lever, 16, mesh with the teeth of the gear segment, 15, the segments being keyed to the shafts, 8. The turning of these shafts affects the movement of the eccentrically pivoted rocking levers, 7, increasing or decreasing the length of the open period of the suction valves according to the direction in which the shafts are turned.

The action of the hand control is shown in Fig. 81, in which the parts bear the same numbers as in Fig. 79. By moving the segment lever, 15, into the position marked "maximum", the suction valve is held open for a shorter period during the instroke of the plunger, 2, and the maximum quantity of fuel is forced through the discharge valve, 12. By moving the segment lever into the position marked "minimum", the suction valve is kept open during the full period of the instroke of the plunger and no delivery takes place past the discharge valve, as all the fuel is forced back into its supply chamber and the engine receives no fuel. Between the "maximum" and "minimum" positions of the segment lever, i.e., between the maximum discharge and no discharge of fuel by

the pump, any intermediate quantity can be delivered to the spray valves to suit the desired engine speed.

In both views of Fig. 81, the positions of the parts are shown at the termination of the in-stroke of the plunger. In view A, the suction valve is still being held from its seat, no delivery of fuel taking place. In view B, the suction valve has been allowed freedom to seat, and its spring has forced it upon its seat at the earliest point in the in-stroke of the plunger.

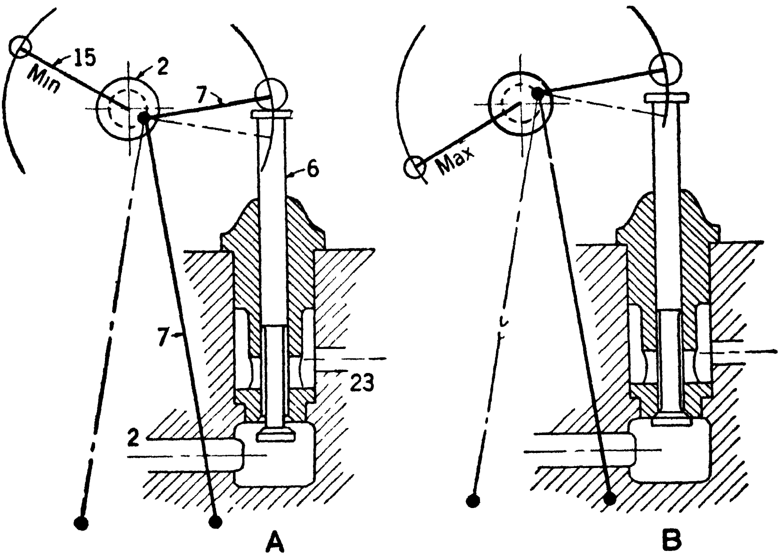


Fig. 81.—Operation of Fuel Pump Suction Valves

The lever, 21, through the rods, 22, is controlled from the overspeed governor; pushing the rods, 22, upwards, forcing the farther end of the levers, 21, downwards upon the rocking levers, 7, entirely cuts off the supply of fuel to all working cylinders by holding open the fuel pump suction valves.

General Electric Diesel Fuel Pump. These fuel pumps were designed to operate at an extremely high speed, in connection with the 150 K.W., 500 R.P.M., 2-cycle opposed piston type Diesel engines built by the General Electric Co. Referring to Fig. 82, A is the crank pin, which actuates the pump plungers through the levers B and C and the yoke D. The

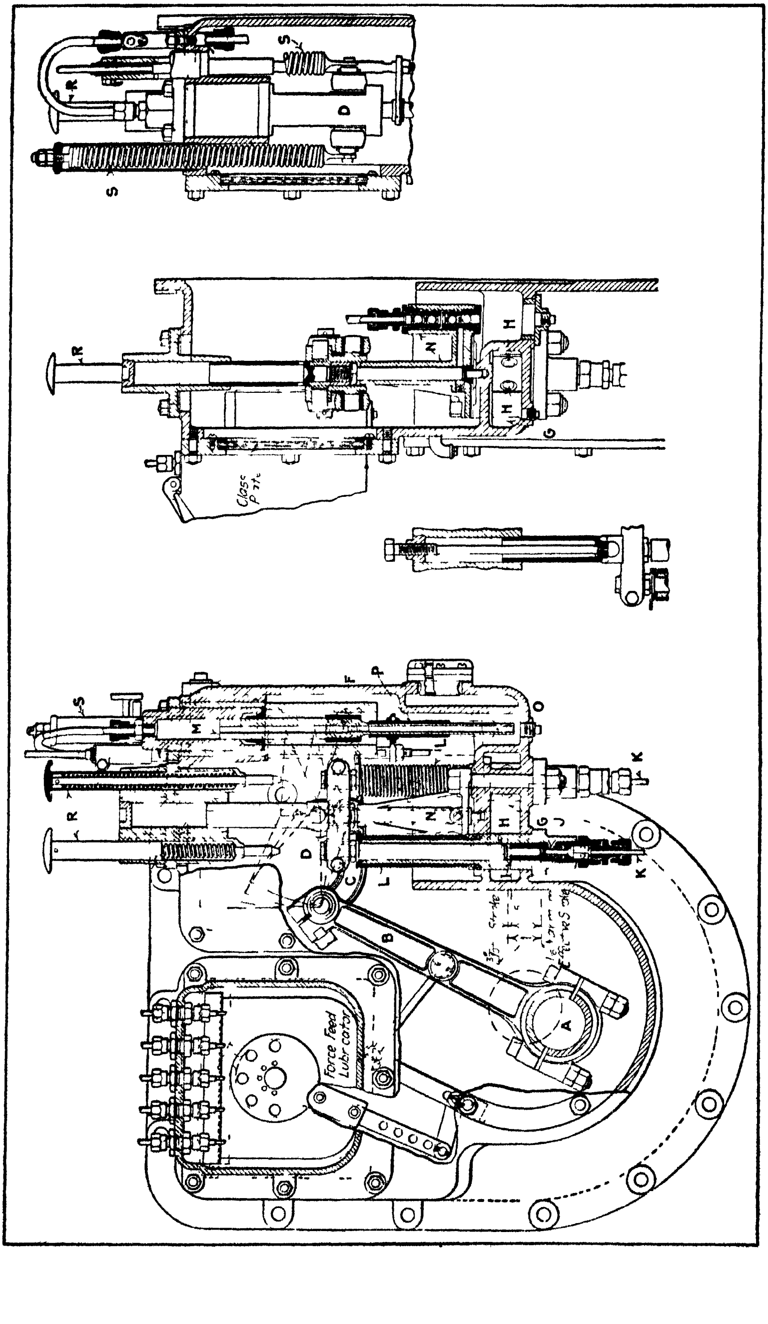


Fig. 82.—Section Through Fuel Pump Manufactured by the General Electric Co.

fuel flows by gravity to the receiver G and through the holes H surrounds the plunger end, which is of tool steel ground to a glass hard finish. The fulcrum F of the lever C is raised and lowered through the action of the governor, and although the stroke is practically constant, the zone of its action varies with the position of the fulcrum F. This is what is known as the "constant stroke variable zone principle". In this way measured quantities of fuel to suit the engine load is pumped by the plunger at I. Two discharge valves in series are provided at J, and the pipes K.K. lead to the two fuel injectors of the engine. The pump cannot become air-bound through low fuel, as any air will rise to the top of the receiver. A removable glass cover on the front of the pump permits its operation to be under the eye of the engineer.

The resistance encountered in forcing out the oil produces a reaction tending to force the fulcrum F upward; this is aided by the springs L.L., whose other function is to return the pump plungers. To increase the quantity of fuel to the spray valves, the fulcrum must be forced down by the plunger, which is under oil pressure from a "regulating pump" N between the fuel pumps and operated by the yoke D. This regulating pump has both suction and discharge valves, and the fuel delivered by it tends to force down the plunger M and the fulcrum F until permitted to escape through a small port O in an extension of M, when uncovered by the end of the sleeve P, which is connected to the governor. When the port is thus uncovered, the pressure is relieved and the reaction of the fuel pumps, aided by the springs causes the fulcrum F to rise until the port O is again covered. A balance is effected in this way depending upon the position of the governor, which is of the centrifugal type. The arrangement relieves the governor of all strain. At no load or very light loads, when the reaction produced by the pump plungers is light, the springs S.S. come into action to raise the fulcrum. By opening a by-pass valve in the regulating-pump delivery line, the pressure on the plunger M is released and the supply of fuel is immediately cut off, due to the fuel pump plungers operating in an non-effective zone.

Worthington Fuel Pump. The Worthington fuel pump as

used on their 2-cycle Diesel engines is shown in Fig. 83. It is of the unpacked type, as can be seen from the drawing. It is driven by the eccentric, 17, mounted on the crankshaft. Each cylinder has a separate and independent pump, complete in all its details. By avoiding the use of packing for the pump

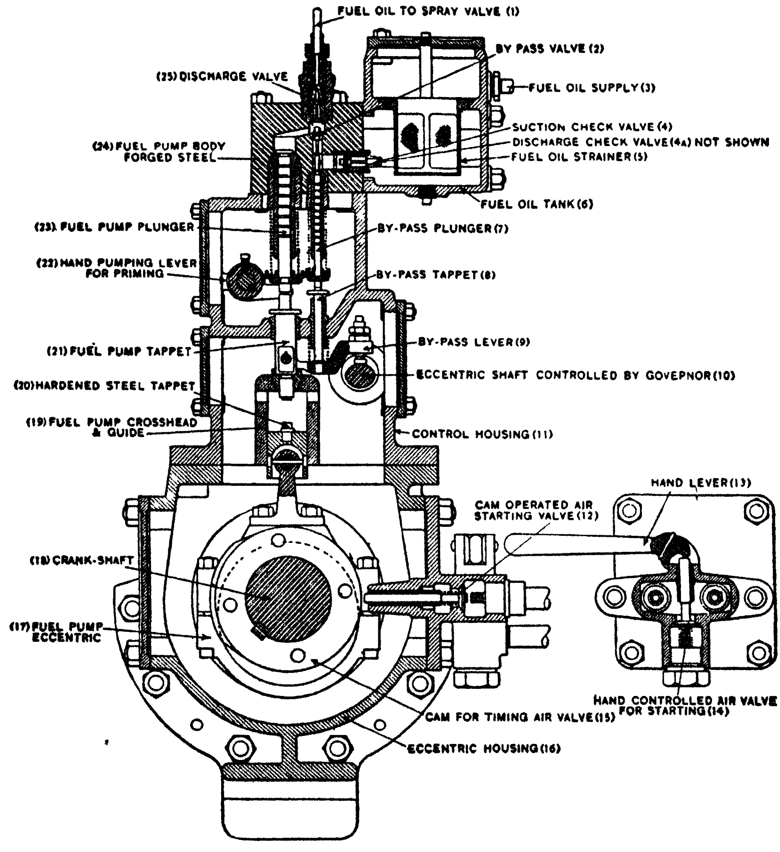


Fig. 83.—Sectional View of Worthington Fuel Pump

plunger, danger of the plunger sticking is eliminated, and at the same time leakage past the plunger is considerably less than in the usual type of packed pump. The pump body, 24, is made from a solid block of steel, designed and constructed with great care, and the plungers and valves are assembled in this body as a unit.

Actuation of the pump plunger is by the eccentric, 17, the crosshead, 19, engaging the tappet, 21, which in turn pushes the plunger upwards on its injection stroke, the plunger being returned to its at rest position by the spring as the eccentric passes its top position. The actual instant at which the pump plunger begins to move upwards determines the instant of injection of fuel into the injection chamber.

For control, a centrifugal governor is fitted, acting through a series of rods and levers to rock the control shaft, 10. This shaft is turned eccentrically at a point in line with each fuel pump plunger, and a so-called "by-pass lever" 9, actuated at one end by the pump plunger, rests at the other end on the eccentric part of the control shaft through the medium of an adjustable set screw. As the end of the by-pass lever moves upward with the pump plunger, the by-pass tappet, 8, and plunger, 7, resting on the lever, will move upwards also, closing the gap between its upper end and the suction or by-pass valve 2. When contact is established at this point, the continued upward movement of the plunger will open the valve, permitting the fuel to by-pass back to the fuel reservoir instead of being forced through the spray valve, and thus ending the injection or effective pump stroke. It will be seen that by rocking the control shaft, 10, one way or the other from its normal running position, the instant of opening of the suction valve by the by-pass plunger may be retarded or advanced, thus increasing or decreasing the effective pump stroke and feeding more or less fuel into the cylinder, according to the load requirements. Also, by regulation of the set screw in the end of the lever, 9, the instant of by-pass opening for each cylinder may be individually adjusted for purposes of initial setting and load balancing.

Pumps That Supply Fuel to Open Type Spray Valves. Fig. 84 shows a cross section of a fuel pump of this type. The pump displacement is in proportion to the travel of the plunger, A, which is slotted, a wedge B, actuated by the governor moving up and down in this slot, decreasing or increasing the length of stroke and the quantity of fuel delivered to the spray valve.

The cam and the oscillating lever or rocking arm, through

which the plunger is actuated, compress a spring, which on its extension reverses the direction of the plunger during the suction stroke of the pump. The plunger can also be operated by the hand lever, before the engine is started, to deliver fuel to the spray valve.

Pumps of this type are of simple construction and are subject to little wear. As they operate under low pressure, the

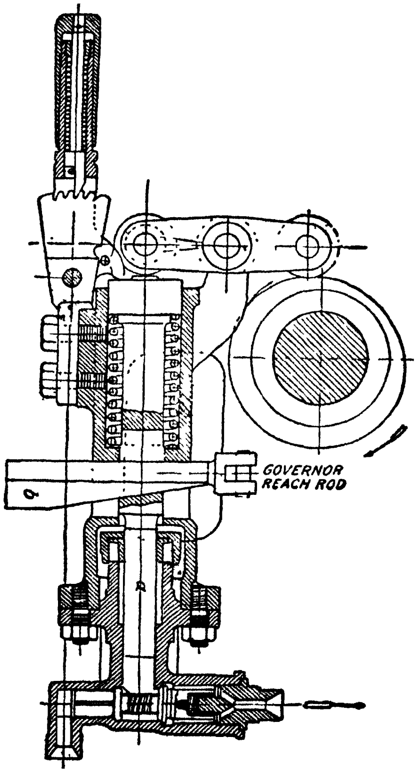


Fig. 84.—Cross Section of Pump Designed to Supply Fuel to Open Nozzles Without Opposing Air Pressure

stuffing boxes are easily kept tight and give little trouble. They are very sensitive to the governor and respond to the slightest load change.

National Transit Fuel Pump. This fuel pump, Fig. 85, is driven by an eccentric on the lay shaft. The plunger is of the differential type, the upper or larger part being hollow and

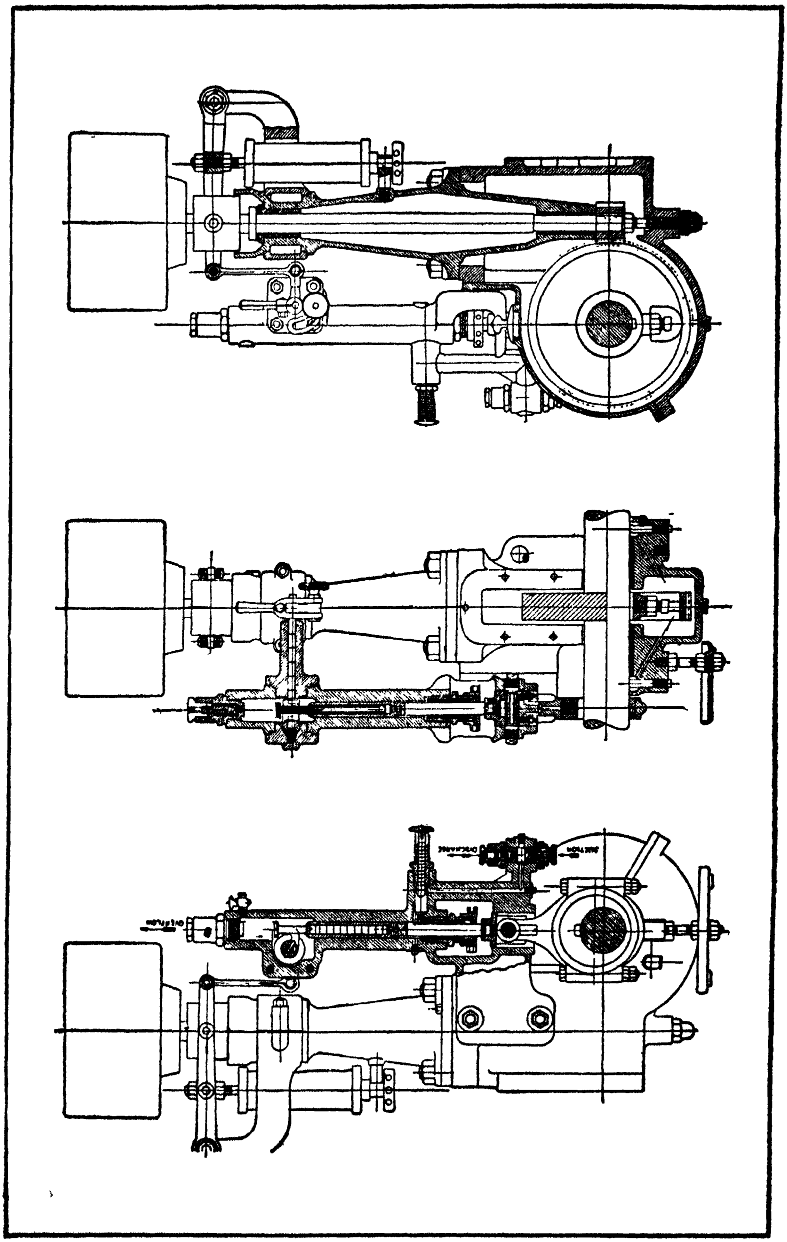


Fig. 85.—Cross Section of National Transit Diesel Fuel Pump

having seated at its upper end a cut-off valve. The plunger has a full positive stroke with each revolution of the lay shaft. the cut-off valve is under the direct control of the governor. Governing is effected by permitting the cut-off valve to seat in a predetermined point in the upstroke of the plunger, thus delivering to the spray valve a quantity of fuel, correctly proportioned to the load that the engine is carrying.

Troubles with Fuel Measuring Pumps. Most of the troubles with fuel pumps are from dirt getting under the suction or discharge valves, or from the check valves leaking and allowing air bubbles to get into the pump chambers and cause them to become air-bound. The plunger packings should not be kept too tight as this will throw unnecessary strain upon the driving mechanism, and they should be so adjusted that the packing leaks a very small amount but not enough to cut down the supply of fuel to the spray valves. The adjusting screws should be kept locked tight or the timing of the pump will be affected. Aside from this, if the pumps are kept clean and properly adjusted they should give but little trouble as they are usually very ruggedly built for the work they have to perform.

A fuel oil strainer and filter should always be installed in the fuel line between the gravity tanks and the fuel pump. This will prevent grit and finely suspended foreign matter in the fuel oil from reaching the pump and causing wear upon the valves and plunger packings.

Adjusting Quantity of Fuel Pumped. As each pump plunger must deliver the same quantity of fuel at each stroke to the spray valves the suction valve lifting mechanism must be properly adjusted to accomplish this. The best method is to take a set of indicator cards and adjust the tappet screws of the lifting mechanism until cards are obtained having the same mean effective pressure. Another method is to measure the fuel discharged by each pump, one at a time, by disconnecting the fuel line leading to the spray valves and allowing the fuel to be discharged into a graduated measure for a given length of time, usually one minute. In either case the engine must run at a given speed and load and without a change in the fuel control lever.

Variable Stroke Fuel Pumps. A diagram of the method of obtaining a variable stroke of pump plunger is shown in Fig. 86. The plunger 1, is driven by the connecting rod 2, which is free to slide in or out along the quadrant 3, its position being governed by the setting of the control wheel 4. The whole mechanism is driven by means of the eccentric 5. By

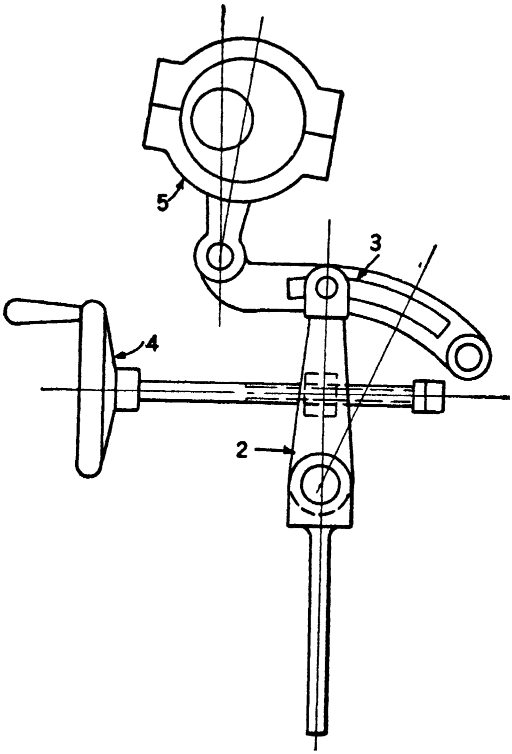


Fig. 86.—Principle of Variable Stroke Fuel Pump

screwing in the control wheel the stroke of the plunger is shortened and by screwing out on the control wheel the plunger stroke is lengthened, the pump then pumping a greater quantity of fuel.

Governors. There are two general types of governors in use for Diesel engines and for entirely different purposes. In the case of stationary engines driving electric generators the

governor is of the sensitive type and takes complete charge of the engine speed and load and governs very accurately to the point set. In this case the governor acts directly upon the fuel pump and there is no hand control other than some way of regulating the setting of the governor.

With marine engines the speed and power of the engine is directly controlled by the hand of the engineer. The hand control acts to change the moment of closing of the fuel pump suction valves, thus controlling the amount of fuel pumped

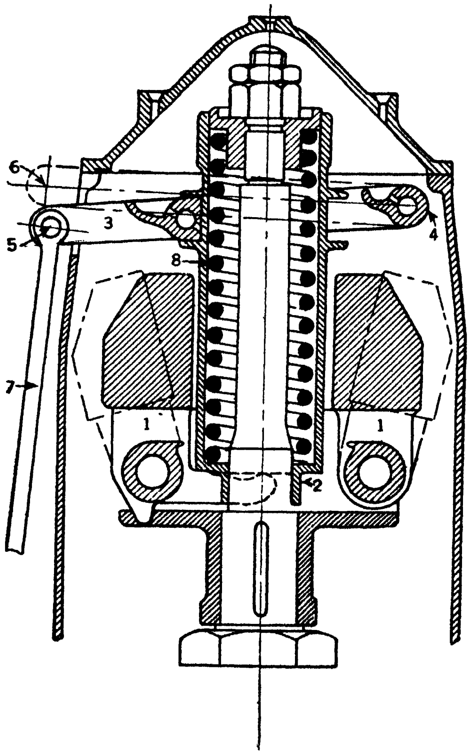


Fig. 87.—Overspeed Governor

and the power of the engine. However, although the speed of the engine is directly under hand control, there are times when the load is suddenly relieved, such as would be the case if the propeller were lost or the clutch became disengaged. If this should happen the engine would run away before the engineer

could stop the engine and serious damage would result. It is for this purpose that overspeed governors are placed on marine engines.

A form of overspeed governor is shown in Fig. 87. When the engine speed rises above normal, the governor weights, 1, fly out into the position indicated by the dotted lines; in so

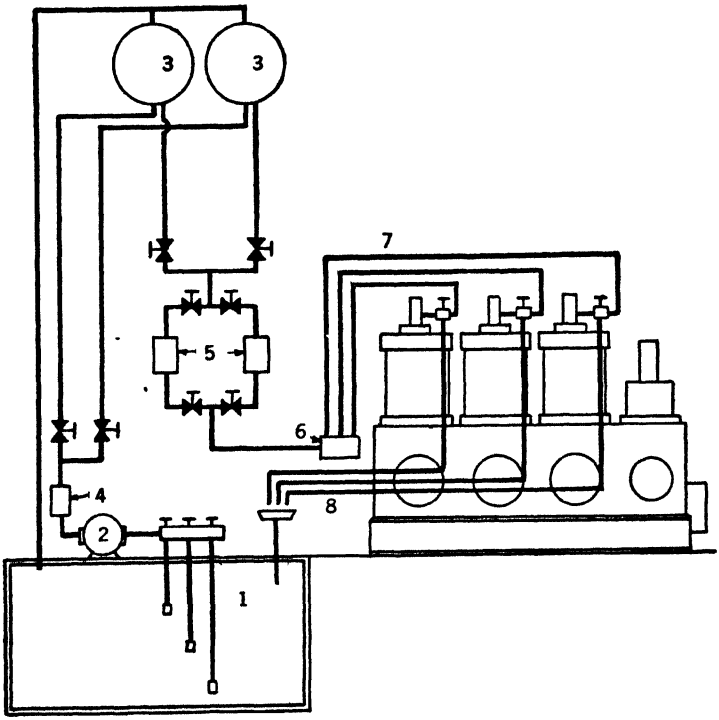


Fig. 88.—Diagrammatic Arrangement of Fuel Oil System

doing they raise the sleeve, 2, and swing the lever, 3, around the fulcrum 4. The pin, 5, is brought into the position, 6, pushing up on the rods, 7, which are equivalent to the rods, 22, Fig. 80 and Fig. 81; thus forcing open the fuel pump suction valves and holding them in this position, which cuts off the supply of fuel to the working cylinders. Upon the engine speed slowing down again to slightly below normal the governor spring, 8, forces the governor weights, 1, into the

closed position, and the fuel supply to the working cylinders is again reestablished.

Lubrication of Governors. The lubrication of governors and governor gears is of great importance. Engines using forced feed lubrication usually have a heavy stream of lubricating oil fed to the governor gears in order to reduce gear teeth cutting. With engines lubricated by hand or gravity feed a very heavy bodied oil must be used for lubrication.

Fuel Oil Systems. Fig. 88 represents diagrammatically a simple layout of the fuel oil system of a small Diesel power plant. There are usually two main fuel oil storage tanks, 1, each containing several months supply of fuel oil. The use of two or more fuel storage tanks provides for a thorough cleaning out of each tank when empty and also a damaged tank can be repaired without causing a shut down of the entire plant. Two or three suction pipes, each fitted with a coarse strainer at their lower ends, are placed at different depths in the tanks so that a suction can always be taken from a high level of the fuel. This is done in order to prevent mechanically held substances, water and impurities from being drawn into the fuel lines. The suction pipes lead to a manifold to which a hand or power driven pump, 2, is connected. This pump supplies the fuel to the two gravity tanks, 3, each of which holds a half day's supply of fuel. On the line between the pump and the gravity tanks is a fuel oil filter, 4. Vent lines lead from the top of the gravity tanks back to the main storage tanks to prevent a loss of fuel in case of an overflow. From the gravity tanks the fuel passes through one of the twin filters, 5, and thence to the fuel measuring pump of the engine, 6. The fuel lines from the measuring pump to the spray valves, and the return by-passed fuel from the fuel check valves back to the main storage tanks are shown at 7 and 8 respectively.

Marine installations are of similar design, with the exception that there are a greater number of fuel storage tanks which are built into the double bottoms of the ship. An ingenious fuel oil system which is installed in the latest United States submarines and known as an automatic fuel oil compensating system is shown in principle in Fig. 89. This

system automatically replaces the fuel used with sea water in order to preserve the trim and diving buoyancy of the boat. A line leading from the main engine circulating water pump, 1, is led forward to a manifold, 2, which has two connections; one leading to the sea and the other to the bottom of the

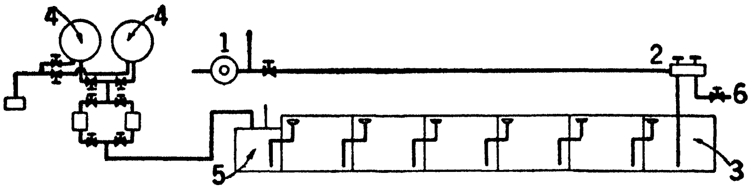


Fig. 89.—Automatic Fuel Oil Compensating System as Used on Some Submarines

forward fuel storage tank, 3. When in operation, a pressure of from five to ten pounds is kept on the forward tank which in turn places all the storage tanks under the pressure of the circulating water system. As the fuel is used by the engine from the gravity tanks, 4, it is automatically replaced by sea water from the circulating system; the water, entering at the bottom of the forward fuel storage tank, floats the fuel aft with the water following. When all the fuel tanks become empty an air pressure is placed on the settling tank, 5, and the water is blown forward from tank to tank and overboard through the sea valve, 6.

CHAPTER VII

Valve Gears—Starting and Reversing Systems

Various Reversing Systems—Valve Actuating Devices—Cams—Camshaft Drives—Rocker Arms—Air Starting Systems—Air Starting Valves —Mechanical Air Starting Systems—Pneumatic Air Starting Systems—Air Systems—Air Storage Systems—High Pressure Air —Reversing Systems in Detail—Reversing Two-Cycle Engines—Reversing Four-Cycle Engines—Reversing by Sliding Camshafts—Reversing by Change of Cams—Reversing by Changing Exhaust to Inlet, and Inlet to Exhaust—Lost Motion Couplings—Pneumatically Operated Reversing Systems.

Valve Actuating Devices. All of the mechanically operated valves on a Diesel engine are opened and closed by some form of valve gear. These usually consist of some combination of rocker arms, push rods or bell cranks which receive their motion from cams. These cams are attached to cam shafts which are driven from the main crankshaft by means of gears.

Fig. 90 is a diagrammatic arrangement of some of the most common methods of driving the cam-shafts of Diesel engines. The spiral gear drive shown in A is one of the most common methods and consists of a vertical shaft having a spiral gear at the bottom and meshing with a similar gear attached to the crankshaft. At the upper end of the vertical drive shaft and meshing with a gear on the camshaft is another spiral gear. The one to two ratio between the crankshaft and the camshaft is usually made with the lower gears. Various combinations of worm, spiral and miter gears with the vertical shaft drive will be found on different makes of engines, the one shown in Fig. 90-A being that of the Worthington engine. At B is shown the Nelseco method of driving the camshafts. These engines have two camshafts, one for the inlet, spray valve and air starting valve, and the other for the exhaust valves. The two camshafts are mounted one on either side of the engine housing, as can be seen by referring to Fig. 195, and are supported in bearings lined with white metal. The timing gears are mounted in a cast housing at the forward end of the engine. The gears on the camshafts are twice the size of the

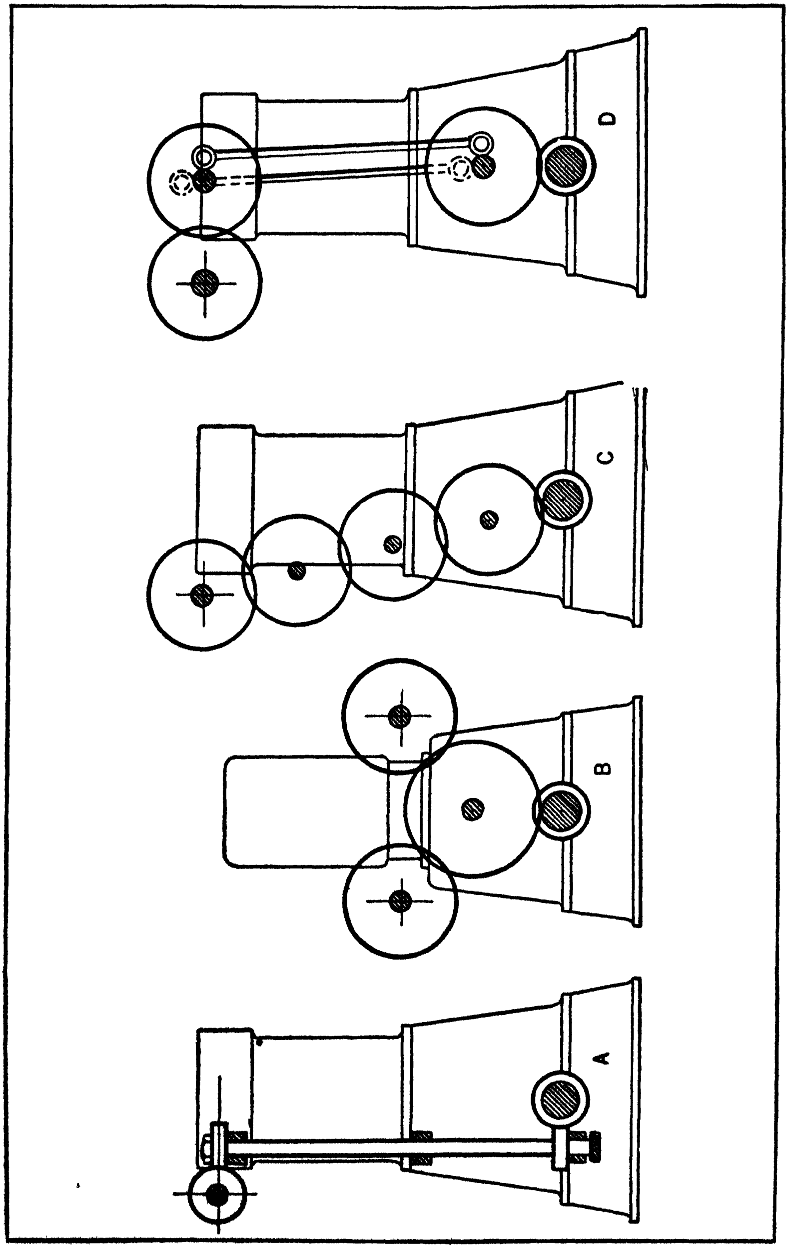


Fig. 90.—Common Methods of Driving the Camshaft

gear on the crankshaft in order to drive the camshaft at half the speed of the crankshaft, the large gear being simply an idler.

At C is shown an all spur gear drive consisting of a chain of spur gears from the crankshaft to the camshaft. This arrangement has certain advantages as the timing is not changed due to the wear of gears, such as sometimes happens with spiral and worm gears. The arrangement shown at D, consists of two pairs of spur gears connected by means of a pair of coupling rods which are driven by a pair of cranks placed at right angles to each other, and is found on some very large engines.

Cams. The cams for operating the various rocker arms and push rods are made of special case hardened steel or chilled cast iron. The cam surface is machined and ground to a master cam. The inlet, exhaust and air starting cams are keyed directly to the cam shaft with no provision for adjustment on the shaft; the spray valve cams, however, are usually made adjustable within a few degrees by means of a removable cam-nose insert which is held in place by screws.

Rocker-Arm Levers. As the inlet and exhaust valves have to be frequently removed for cleaning and grinding, it is desirable that they be readily removed without disturbing the valve operating gear and cam clearances. This is accomplished by the construction shown in Fig. 91. The rocker arm that engages the valve is split and the two parts are joined with two dowell pins and held together by a bolt. By removing the bolt and a part of the rocker arm, the entire valve and cage can be removed without disturbing the rest of the valve gear.

Air Starting Systems. At present the majority of Diesel engines require compressed air for starting, although the method in which the air is used differs, according to the design of the engine.

To start a Diesel engine, it is, of course, necessary to make the shaft revolve, forcing the pistons to compress the air in the cylinders sufficiently to ensure combustion, when the fuel charge is injected. The air starting gear is then thrown out of action. The operation of starting can usually be completed in three or four revolutions of the crankshaft.

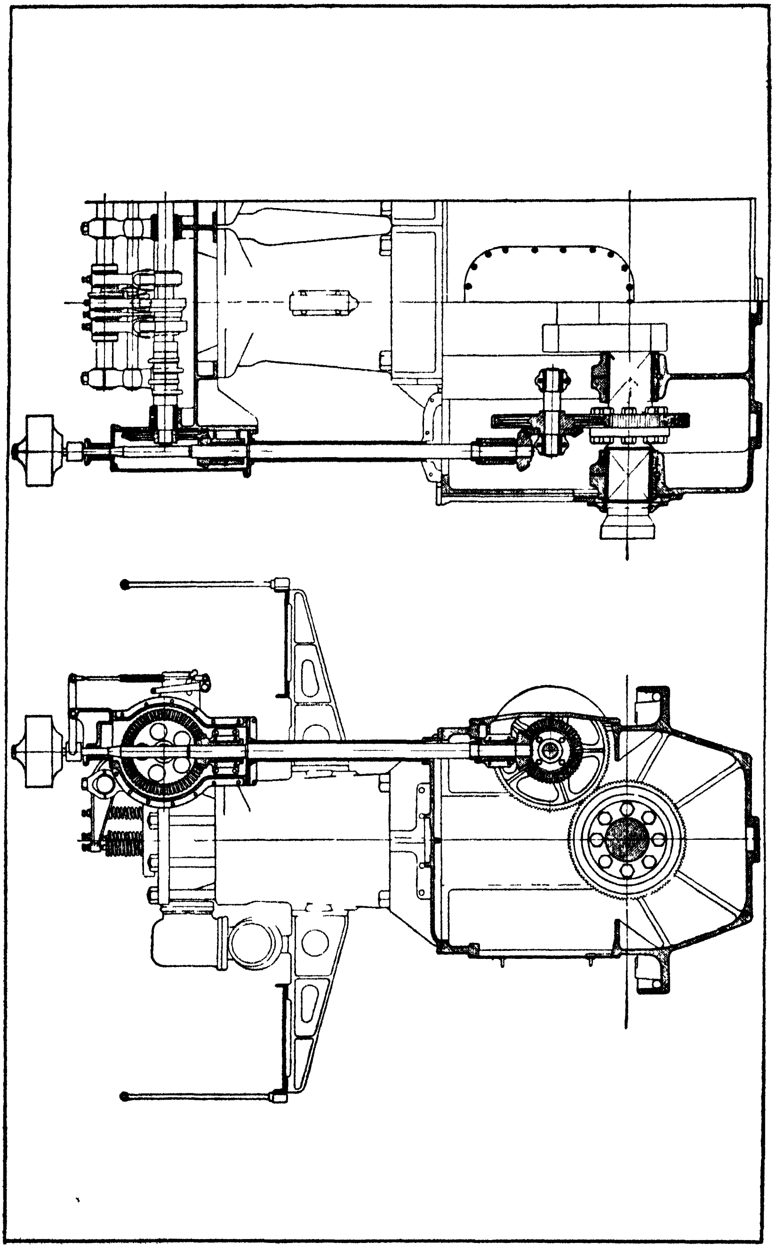


Fig. 90a.—Method of Driving Camshaft of Worthington 4-Cycle Engines

There are two distinct methods of air starting and the air starting valve is operated in either one of the two following ways

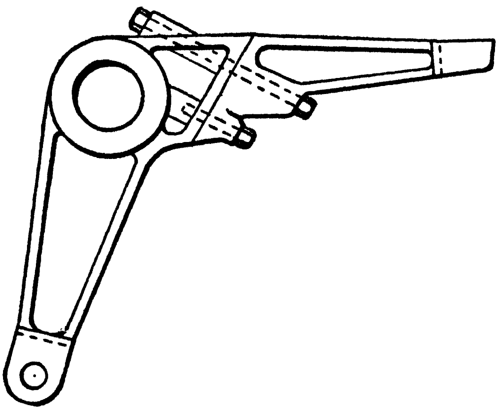


Fig. 91.—Adjustable Rocker Arm Lever

(A) Mechanically operated. By which the air starting valve is operated by means of cams and levers.

(B) Pneumatically operated. By which air is admitted behind a piston of the air starting valve, thus forcing the valve open.

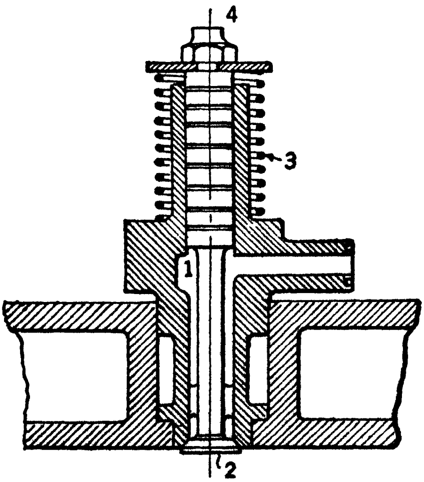


Fig. 92.—Mechanically Operated Air Starting Valve

In either system the air may be made to act on one or all of the working cylinders or, in the case of the step-piston type of 2-cycle engine, the air may act directly on the scavenging piston, the working cylinders being thrown immediately on to fuel at the start.

Mechanical Air Starting Systems. A typical mechanically operated air starting valve is shown in Fig. 92. The valve and

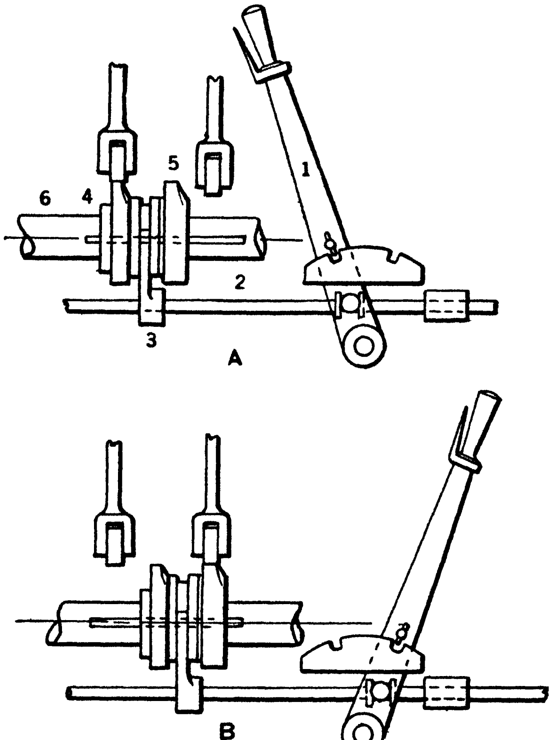


Fig. 93.—Mechanically Operated Air Starting Arrangement

cage fit into a pocket in the cylinder head. Air is admitted from the air starting bottles into the annular space, 1, where its pressure on the top of the valve, 2, is balanced by the pressure on the enlarged portion of the valve spindle; the force tending to keep the valve on its seat being supplied by the spring, 3. At the proper instant (about 10 to 15 degrees past top center on the working stroke) a cam engages the air start-

ing valve rocker arm which strikes the tapper nut, 4, and thus opens the valve. High pressure air of from 300 to 800 pounds pressure per square inch is then admitted to the cylinder and forces the piston downward. The valve is allowed to close at from 70 to 90 degrees past top center and the air then expands during the remainder of the piston stroke.

One of the simplest air starting arrangements is shown diagrammatically for one cylinder in Fig. 93. With this type of starting gear the usual practice is to have only half the number of working cylinders equipped for air starting. As the period of opening of the air starting valves do not overlap it is necessary to jack the engine over until one of the starting valves have just opened, the hand lever, 1, having previously been thrown into the starting position as shown in A. Attached to the hand lever is a long rod, 2, carrying a forked collar, 3, which causes the air starting cam, 4, and spray valve cam, 5, to be shifted to a position under either one of the two push rods. The two cams rotate with the camshaft, 6, but are free to slide back and forth on a key.

On a four cylinder engine the two central cylinders with cranks placed 180 degrees apart will usually be found equipped for air starting. In operation, the other two cylinders are made ready for firing. The fuel lines are primed and the spray air flask charged with air at a pressure of about 700 pounds per square inch. The engine is then jacked over until one of the starting valves has just opened, then, by opening the air starting throttle valve the engine will start to turn over and, as soon as the other two cylinders start to fire on fuel, the hand lever is thrown back into the position shown at B, removing the air starting cam from under the air starting valve push rod and bringing the spray valve cam under the spray valve push rod. All cylinders will then start firing on fuel. This method of air starting is only found on small stationary engines and on marine engines equipped with reversing clutches.

Fig. 94 shows the general arrangement of the air starting and spray valve actuating gear for most 4-cycle Diesel engines. The air starting valve, 3, is actuated by the rocker arm 1, while the spray valve, 4, is actuated by the rocker arm 2. Both of

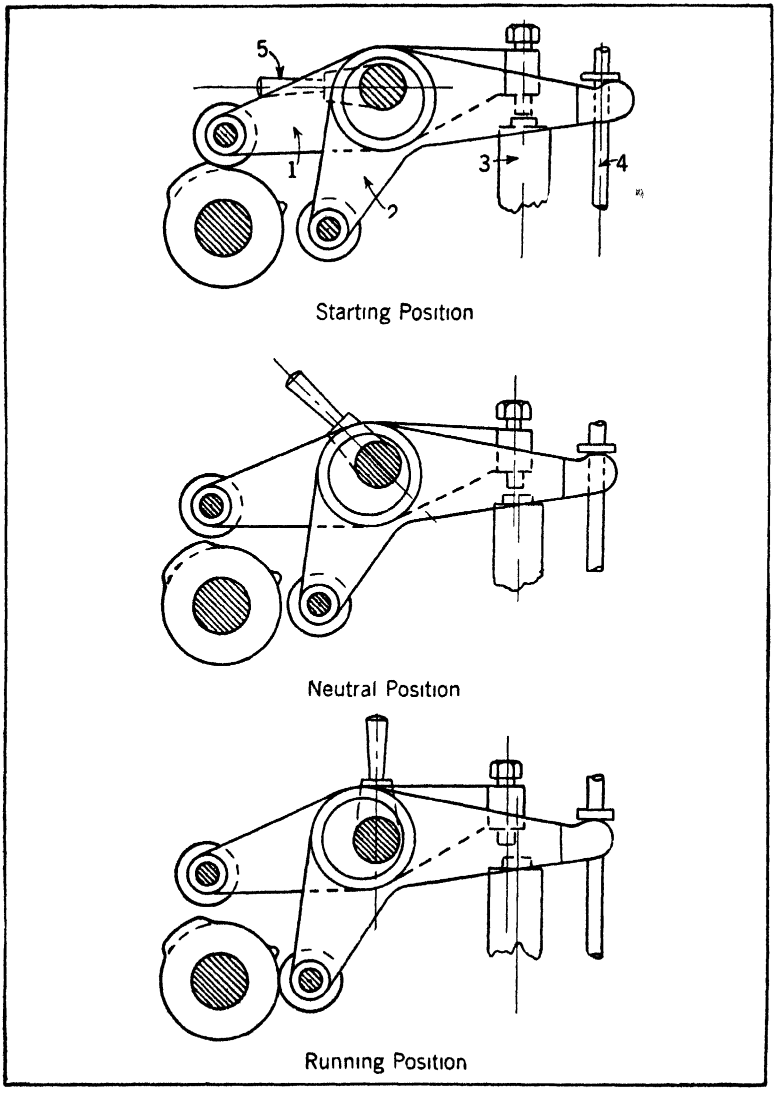


Fig. 94.—Method of Cutting in the Air Starting and Spray Valve Operating Mechanism

these rocker arms are mounted on eccentrics which can be moved through an angle of 90 degrees by means of the hand lever, 5. As shown in the starting position, the air starting valve rocker arm, 1, is shown in contact with its cam, and the spray valve rocker arm is thrown out of contact with its cam. By moving the hand lever to the neutral position both rocker arms are placed out of contact with their respective cams. By shifting the hand lever still farther into the running position

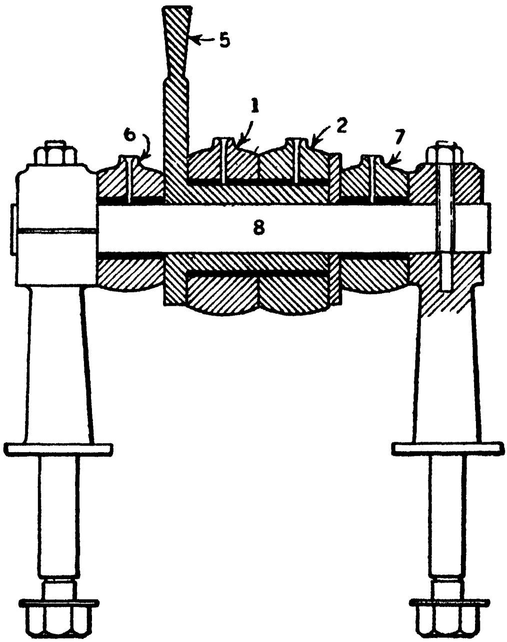


Fig. 95.—Arrangement of Rocker Arms of Valve Gear Shown in Fig. 94

the spray valve rocker arm is placed into operation and the air starting rocker is moved out of range with its cam.

The general arrangement of this form of valve actuating gear is shown in Fig. 95. Besides the air starting valve rocker arm, 1, and the spray valve rocker arm, 2, which are both mounted on the eccentric moved by the hand lever 5, are shown the inlet valve rocker arm, 6, and the exhaust valve rocker arm, 7. All of these rocker arms are mounted on the fixed shaft, 8,

which acts as a fulcrum for the rocker arms. This shaft is supported between two columns which are mounted on the cylinder head or attached to the engine framing.

With some large multi-cylinder engines the shift lever, 5, of all the working cylinders is operated by one lever from the operating platform. Some engines have two such levers, each lever operating the shifting gear of one half of the number of working cylinders. This quickly permits half the number of cylinders to operate on air while the others are made to fire

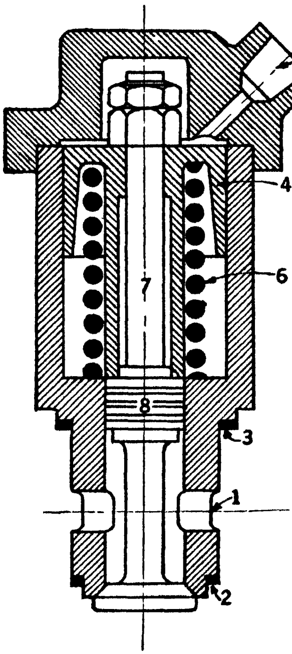


Fig. 96.—Pneumatically Operated Air Starting Valve

on fuel. For stationary engines many designers provide this starting gear for only two cylinders, or for even only one cylinder for engines having less than four cylinders.

Where the hand shift levers are not connected and brought to one central point of control, the method of starting is to first place all cylinders in the air starting position and then open the air starting throttle valve. The different cylinders are then put into action one at a time, commencing with the one

that experience has shown to be the readiest starter, the others continuing to be operated by air.

Pneumatic Air Starting Systems. Fig. 96 shows the construction of a typical pneumatically operated air starting valve. These valves, instead of being opened by mechanical means, are caused to open by the pressure of air acting on a piston to which the valve and stem are attached. The main air starting supply from the starting bottles is led in through the cylinder head into the valve cage openings, 1. This air is prevented from escaping into the cylinder by a gasket, 2, and from around the top of the valve cage by another gasket, 3. Attached to the top of the valve stem is a large piston, 4, which by means of a heavy spring holds the valve tightly to its seat. At the proper time for admitting starting air into the cylinder the valve is caused to open by admitting actuating air on top of the piston, the air entering through the passage, 5. The area of the piston and pressure of the actuating air is sufficient to overcome the action of the spring, 6, while the pressure of the starting air tending to open the valve, 7, is just balanced by the enlarged portion of the valve stem, 8, which is made in the form of a piston and has several piston rings to prevent the escape of air.

The method of admitting the actuating air at the proper instant is by a distributer, the essentials of which are shown diagrammatically in Fig. 97. The distributer consists of six air valves, V, one being for each cylinder. Below these air valves is a cam, C, which is operated by the camshaft. When air is turned on the six valves are driven down by the air against the action of springs and all are closed, with the exception of one valve which strikes on top of the cam. This valve being held open by the cam, permits the air to pass to the air starting valve of the cylinder and opens the valve thus permitting the starting air to enter the cylinder and act upon the working piston. The engine then turns over and the cam likewise turns, opening the different valves in succession. The engine is reversed by first stopping the engine and shifting the cam, C, to its opposite position and the air stop valve is then opened. The engine and cam then turns in the opposite direction and opens the valves in the reverse order.

This form of air starting arrangement is found on the Ingersoll-Rand engine. With this engine the valve timing and action of the fuel pumps are so designed as to be perfectly symmetrical with respect to the dead center, so that

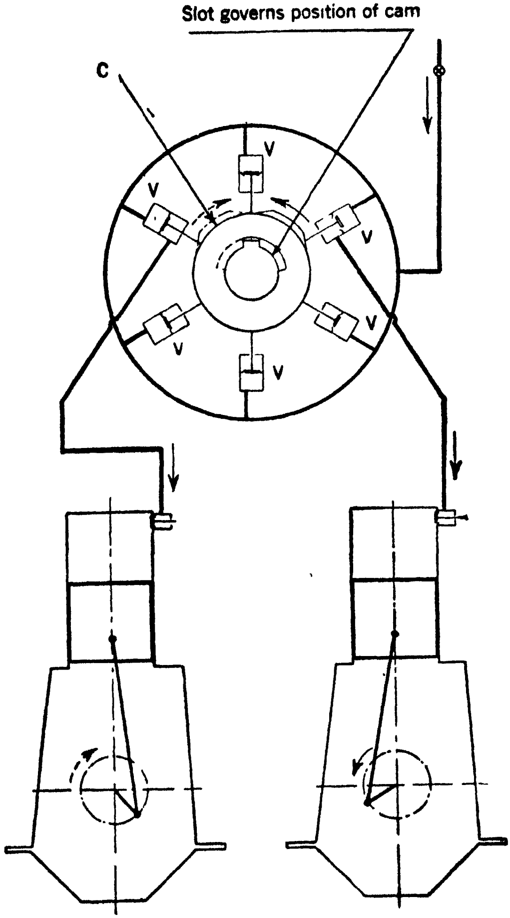


Fig. 97.—Pneumatically Operated Air Starting Arrangement

no matter which direction the engine rotates the valve timing will be identical, with the exception that the exhaust valve will become the intake valve and the intake valve will become the exhaust valve.

Fig. 98 shows diagrammatically the pneumatic air starting arrangement found on some of the Nelseco engines installed in American submarines. With this arrangement, by turning the control wheel, actuating air is admitted first to two, then to four, and then to six working cylinders in succession. When the engine is open up to firing speed, by reversing the motion of the control wheel two cylinders are automatically cut out from the air starting condition and made to operate on fuel. By further turning the control wheel two more cylinders are changed from air to fuel and finally all cylinders are operated on fuel. This is accomplished as follows:

At A is shown a group of six cam valves which are operated by the air starting cam attached to the camshaft. When the depressed part of the cam comes under one of the valve rollers, compressed air will force the valve piston inwards against the action of a spring and uncover a port as shown in cam valve 1, provided, of course, that there is an air pressure in the line. This air will then pass through the port and up to the pneumatically operated air starting valve on the working cylinder, opening the valve and admitting starting air into the working cylinder. As the engine turns over the cam causes the valve to close, and as it does so, the piston will uncover a vent as shown in cam valve 6. This allows the escape of the air holding open the air starting valve on the cylinder and causing it to close. As the engine rotates each cam valve is opened and closed in succession, allowing the passage of actuating air to the air starting valves and opening these valves, then venting the pressure as the cam valve is closed by the cam, causing the release of the actuating air and closing of the air starting valves.

At B, C and D are shown the control relay valves. When these relay valves are closed no actuating air is admitted behind the pistons of the cam valves and they do not operate. Each relay valve admits actuating air to the cam valves in groups of two, and the relay valves are opened by turning the control wheel, E. As the control wheel is turned a cam will first lift the relay valve B, while relay valves C and D are still closed. This admits actuating air to cam valves 1 and 6. Turning the control still farther will cause another cam

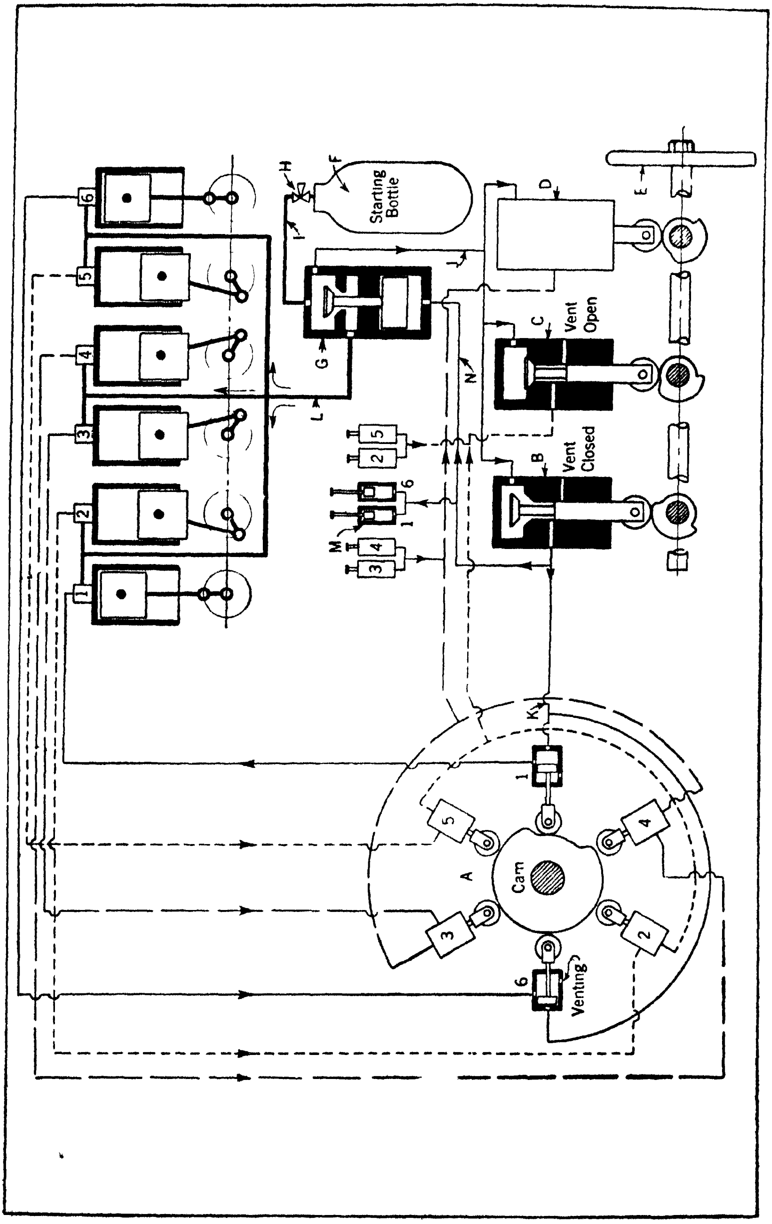


Fig. 98.—Pneumatically Operated Air Starting Arrangement of Nelseco Submarine Engines

to lift relay valve C. Actuating air will now be admitted to cam valves 1 and 6, and 2 and 5. Finally, when the control wheel is turned to full air start, all the relay valves will be opened and actuating air will be admitted to all six cam valves.

When this actuating air is admitted to the cam valves it is also admitted to the fuel pump suction valve cut-outs, M. These cut-outs consist of small cylinders and pistons which hold open the suction valves of the fuel measuring pumps and thus prevents the delivery of fuel to the working cylinders while they are operating on air.

At F is shown the starting air bottle from which air is led to the top of the main relay valve, G. When the stop valve, H, is opened, air passes through the line, I, into the relay and out again through the line, J, which leads to the top of each relay valve, B, C and D. When the relay valve, B, is opened air will pass through the line, N, and act on the piston of the main relay, thus opening the valve and allowing the main body of starting air to pass out through the line, L, to the air starting valves of the working cylinders.

In operation, the control wheel, E, is turned to the full air starting position, lifting all of the relay valves which in turn supply air to all cam valves, the fuel pump suction valve cut outs, and opens the main relay valve, G. As the depressed part of the cam will be in a position to allow the opening of one of the cam valves, the air will pass through and actuate the air starting valve of that cylinder. This cam valve will be the proper one to supply air to the working cylinder in position to receive starting air, i.e., between 15 and 75 degrees on what would be the combustion stroke if the cylinder were firing on fuel.

All six cylinders will now be operating on air. By turning the control wheel, E, back one notch the relay valve, D, will close. When this relay valve closes the valve stem will uncover a vent (see relay valve C) and allow the release of the air pressure from cam valves 3 and 4 and the lines leading to them. This also releases the pressure from the fuel pump cut-outs, allowing the fuel pumps of No. 3 and 4 cylinders to function and deliver fuel to these working cylinders. These two cylinders will thus operate on fuel while cylinders 1 and

6, and 2 and 5, will continue to operate on air. When the control wheel is turned back one more notch, relay valve C is closed and the venting of cam valves 2 and 5 and fuel pump cut-outs for these two cylinders is accomplished. Four cylinders will now be firing on fuel while cylinders 1 and 6 continue to be operated by air. Finally, by turning the control wheel to the shut off position, relay valve B is closed. When this relay valve closes, in addition to venting the cam valves and fuel pump cut-outs for No. 1 and 6 cylinders, the air is

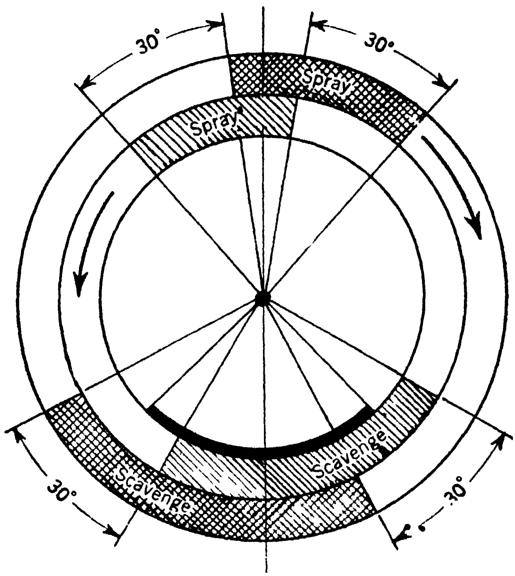


Fig. 99.—Valve Timing of Overhead Valve Scavenging 2-Cycle Engine

released from under the piston of the main relay valve, G, causing this valve to close and cut off the supply of starting air to the air starting valves of all the working cylinders.

Reversing of Two-Cycle Engines. When an engine is running in an ahead direction a definite sequence of events takes place in the cylinders. To reverse an engine, therefore, it is necessary to reverse this sequence in relation to the crankshaft. This sequence is controlled by the cams of the camshaft, so in order to reverse the engine some means must be

adopted to readjust the position of the cams in relation to the crankshaft.

Fig. 99 shows a diagram of the opening of the spray and scavenging valves of an overhead valve scavenging 2-cycle engine, and also the period of exhaust port opening. On the outside of the circle are seen the periods of opening of the spray and scavenging valves when the engine is running in an ahead direction and on the inside of the circle is shown the same period of opening when the engine is running in the astern direction. The exhaust ports, of course, will be opened and closed by the piston at the same time regardless of the direction of rotation. The difference in these valve openings is about 30 degrees, so in order to reverse this type of engine it becomes only necessary to change the timing of the spray and scavenging valves about 30 degrees. This change is accomplished in several different ways.

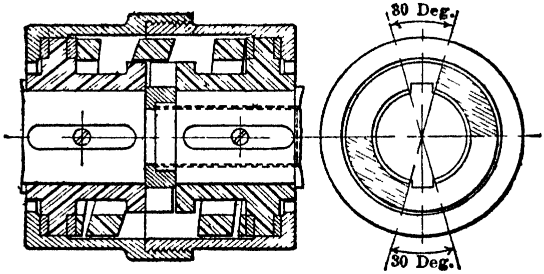


Fig. 100.—Lost Motion Coupling

Reversing by Means of Lost Motion Coupling. The simplest means of shifting the camshaft and its attached cams is by a lost motion coupling such as shown in Fig. 100. This coupling has a 30 degree angle between the jaws, so that when the engine starts in the reverse direction the crankshaft revolves 30 degrees before the camshaft starts to turn and thus delays the valve timing the right amount for reverse rotation. This method is unsatisfactory when an engine is operated at low speeds, as the engine does not turn uniformly and causes a jerky motion to be transmitted to the camshaft. As the camshaft and its attached cams have an inertia of

their own the coupling fails to hold the camshaft in exact relation to the crankshaft; this changes the timing of the valves which further aggravates the uneven running of the engine.

This method has been improved upon by obtaining the proper adjustment of the camshaft by spiral gears. Owing to the spiral angle of the teeth of the gears, if the vertical shaft is moved axially, a definite angular displacement must take place between the crankshaft and the camshaft, thus making the required change in timing of about 30 degrees.

Reversing by Changing to Separate Set of Cams. One of the most satisfactory systems is that having two sets of cams; one for the ahead rotation and one set for astern rotation. The rocker arm which operates the spray and scavenging valves have two sets of rollers as shown in Fig. 101. The operation of reversing consists simply of shifting the rocker

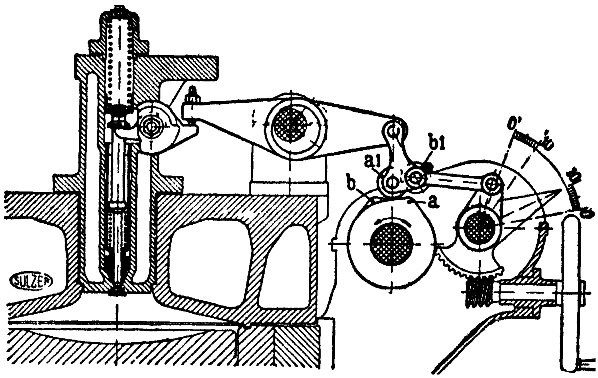


Fig. 101.—Reversing by Changing from One Set of Cams to Another

arms and changing from one set of cams to the other set. This system further provides for an adjusting range of about 10 degrees by which the injection of fuel can be controlled, thus insuring regular firing at low speeds.

Another method of reversing used on both two and four cycle engines is by having two camshafts mounted on a swinging frame such as shown simply in Fig. 102. One camshaft has all the various cams set for ahead running and the other set for astern running. Reversing is accomplished by swinging the frame which carries the two camshafts so as to bring either the ahead or astern cams under the rocker arm rollers.

Reversing of Four-Cycle Engines. The reversal of 4-cycle engines is somewhat more difficult than that of the 2-cycle type. This is principally on account of the different degree changes necessary in rearranging the various cams for astern

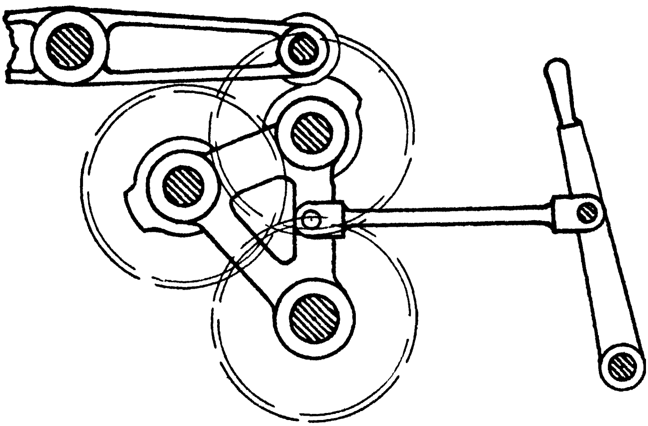


Fig. 102.—Method of Reversing by Changing Camshafts

rotation. A great number of different gears and arrangements for reversing 4-cycle engines have been patented, but in actual use only about three different methods, with slight variations, will be found. These different methods are as follows.

Reversing by Means of Sliding Camshafts. This method has found the most favor with designers of 4-cycle engines. The engine camshaft is equipped with two sets of cams side by side; one set being placed for ahead running and the other set for astern running. Reversal is accomplished by first lifting all cam rollers clear of the cams in order to prevent fouling and then the cam shaft is moved axially a few inches, sufficient to change from one set of cams to the other. The cam rollers are then brought into contact with the cams again.

Fig. 103 shows the elements of such a reversing system. At 1 is shown a compressed air cylinder and piston, the piston of which is caused to travel from one end of the cylinder to the other by means of the reversing cock, 2. The oil cylinder and piston, E, acts as a buffer and prevents a too rapid travel

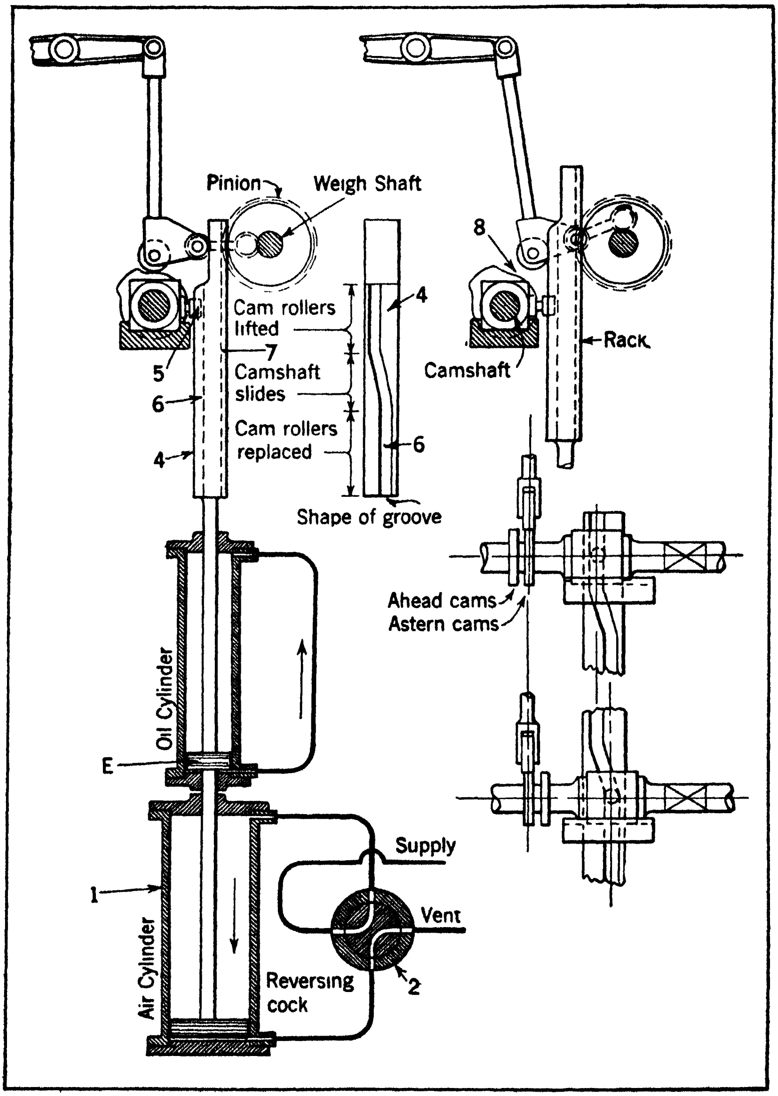


Fig. 103.—Typical Method of Reversing 4-Cycle Engines by Sliding Cam-Shaft and Changing from One Set of Cams to Another Set

of the air piston, the oil being forced through a small pipe from one side of the piston to the other side. At 4 is shown the camshaft actuating slide, consisting of a long groove on one side into which the roller, 5, attached to the camshaft slide, 6, travels. On the opposite side is a toothed rack which meshes with the pinion gear, 7. As the air piston is forced upwards the rack first turns the pinion and the weigh shaft which is fastened to it. To this weigh shaft are connected all the push rods and cam rollers by means of connecting links. During the first quarter revolution of the weigh shaft all of the push rods and cam rollers are pulled clear of the camshaft as shown at 8. During the next half revolution

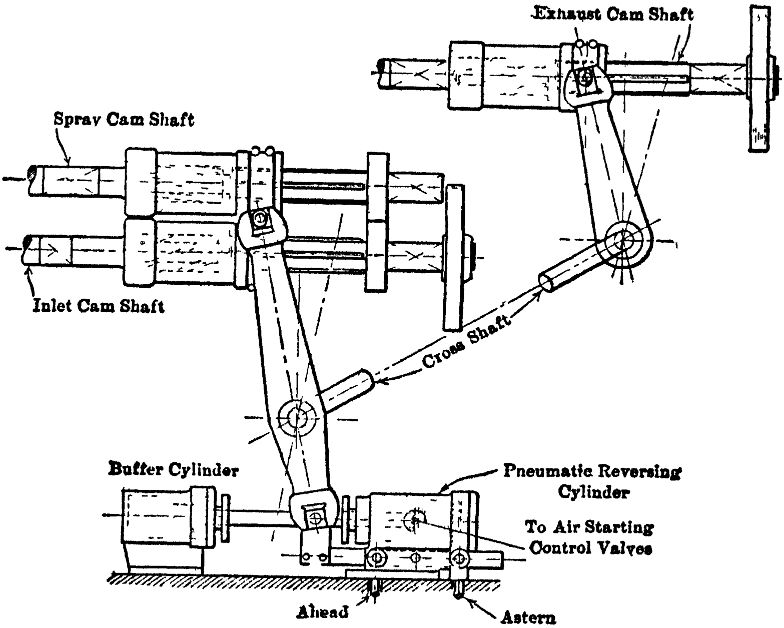


Fig. 104.—Reversing Arrangement of Nelseco Submarine Diesel Engine

the groove in the slide forces the camshaft lengthwise in its bearings, sufficient to change from one set of cams to the other. The next quarter revolution returns the cam rollers and push rods to their operating position over the cams.

With some very small engines the necessity of removing the cam rollers from the cams is overcome by having curved

faced cams and rollers so designed as to slide from one cam to the other.

Reversing by Causing a Degree Change in the Camshaft in Relation to the Crankshaft. This arrangement is necessarily used with engines having separate camshafts for the spray, inlet and exhaust valves, each camshaft which must be changed a different number of degrees in relation to the crankshaft. The three camshafts are shown in Fig. 104 together with their timing gears. The timing gears remain in mesh at all times and are not altered in any way while reversing. Between each timing gear and its camshaft is a reversing clutch which may be slid endwise on a splined shaft to which the timing gears are attached. On the opposite end of the clutch are cut spiral slots which engage the camshaft. As the piston of the pneumatic reversing cylinder is moved forward the reversing clutches are moved also by levers as shown. This slides the reversing clutches endwise on the splined shaft and the spiral grooves in the opposite ends will cause the necessary change in the timing for going from ahead to astern.

Reversal by Change of Valves. In this system a water cooled manifold is used for both the inlet and exhaust. The inlet and exhaust valves are simply changed for reversing, that is, the inlet manifold and valves become the exhaust and the exhaust valves and manifold become the inlet; the only change in timing being the spray and air starting valves which are usually changed by means of a lost motion coupling or clutch.

CHAPTER VIII

Lubricating and Circulating Water Systems

Methods of Lubrication—Forced Feed—Gravity Feed—Sump Tanks—Settling Tanks—Regulating the Distribution of Oil to Main Bearings—Effect of Large Main Bearing Clearances—Check Valves in Connecting Rods—Cylindrical Lubrication—Lubrication Fittings—Mechanical Lubricators—Centrifugal Banjo Lubricators—Lubrication of Gears and Cams—Lubrication of Air Compressors—Troubles with Water or Carbon in the Oil—Oil Coolers—Oil Cleaners—Circulating Water Systems—Volume of Water Required—Path of Cooling Water—Dangers of Excessive Cooling—Cooling of Valves—Cleaning of Sediment from Cylinders and Jackets—Circulating Pumps.

Methods of Lubrication. There are two general systems in use for the lubrication of the main bearings, crank bearings and wrist pins. The first, and most common system, is that of forced feed of the lubricating oil. Other systems use some form of gravity feed or centrifugal oiling device, but this system is only used in the smallest installations.

With the forced feed system the oil is used over and over again. The oil is supplied by a rotary or plunger pump, driven either by the engine or independently. The pump receives its supply of oil from the sump tank and discharges it through a strainer and cooler to a distributing line, which may be a closed channel in the bed plate or a pipe running alongside the engine from which leads are provided to each main bearing. A typical installation is shown diagrammatically in Fig. 105.

From the main bearings the oil is forced to the crank pin and wrist pin bearing as shown in Fig. 106. This system effectively lubricates the wrist pin bearing, which is of particular importance, as its position inside the trunk piston and in close proximity to the piston heads, subjects it to heat by absorption and conduction, in addition to the severe duty it has to perform regularly in transmitting the high piston pressures.

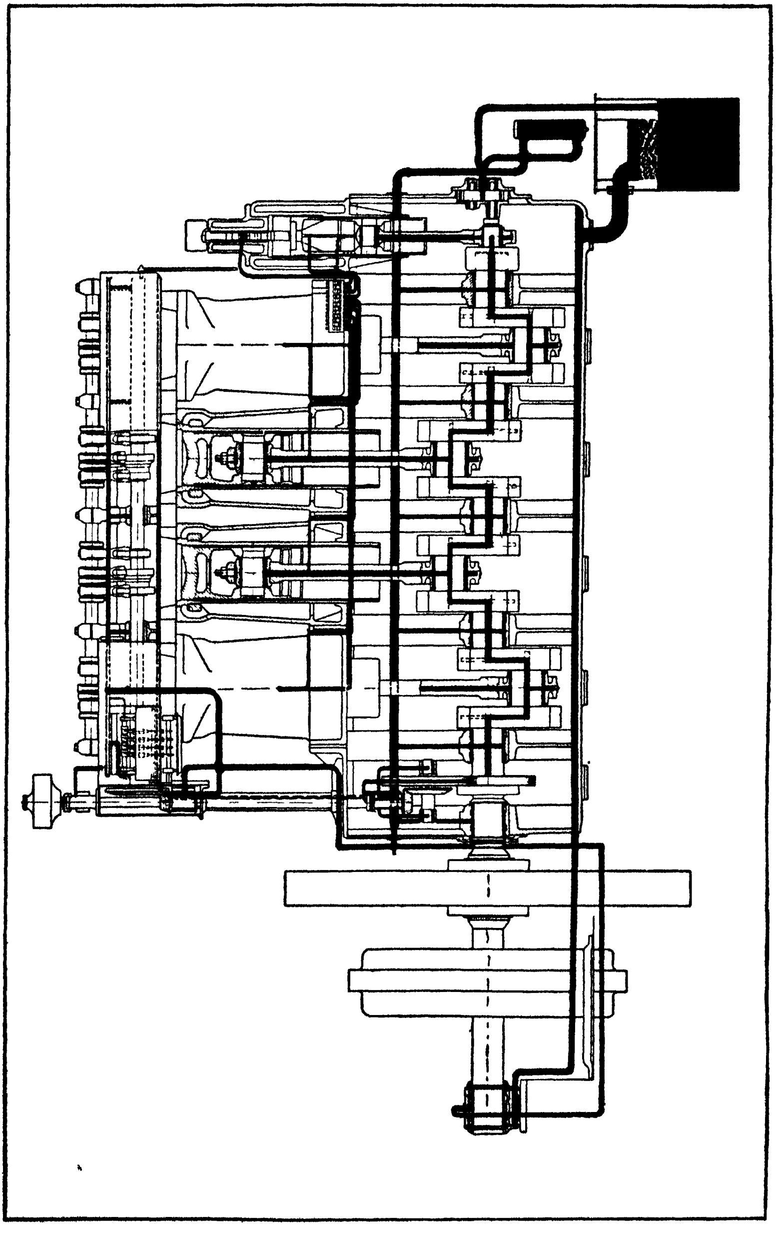


Fig. 105.—Path of Lubricating Oil Through Stationary Engine

The end of the suction pipe in the sump tank is usually fitted with a strainer to prevent entry of rags and waste or other foreign matter into the suction pipe. A second strainer is usually placed on the discharge side of the pump. This strainer is always of the duplex type, so that one side can be cleaned while the other side is in operation. The strainer basket is usually of finer wire mesh than the suction strainer

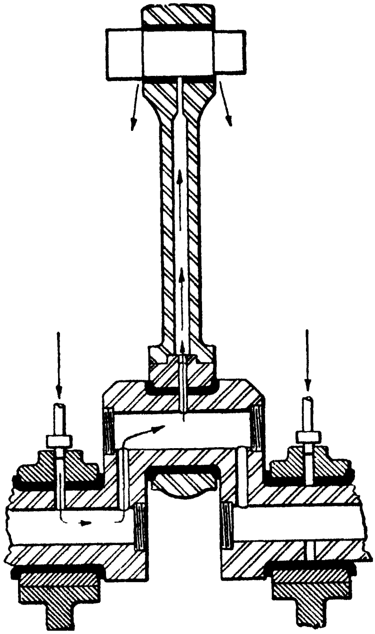


Fig. 106.—Path of Lubricating Oil Through Crankshaft and Connecting Rod to Wrist Pin

and is made removable so as to be easily removed for cleaning.

In marine use the sump tank forms a reserve in the system, and is supplied as needed from the main lubricating oil tanks. The tank is placed at one end of the engine below the level of the crank case and the drains from the crank case should be so placed that the oil will not bank up due to the trim of the ship. One sump tank may serve both engines, but the usual practice is to have one sump for each engine, with cross connections to either engine. A settling tank is usually fitted inside of the sump tank and the oil from the crank case

drains first into the settling tank and then overflows into the sump. This allows very fine foreign matter to settle out, which otherwise would pass through the filters and strainers. Drains should be provided in the settling tank for drawing off water and sediment.

Enclosed Crank Cases Necessary With Forced Feed. Since one of the characteristics of the forced feed system is the circulation of much more oil than necessary for lubrication alone, in order to cool the bearings somewhat, excess oil is thrown from the rotating parts and sprays around in such a way that a tightly enclosed crank case is very necessary to prevent wastage of oil. Another and even more important reason for tightly enclosed crankcases is to keep dust and dirt from entering the lubricating system where they would form an abrasive and cause rapid bearing wear.

The oil sprayed from the crank and crankshaft bearings is sufficient to reach the cylinders to a considerable extent in trunk piston engines, and in some cases it has been found to be ample for the lubrication of the cylinders. In fact, it is difficult to prevent too much oil from reaching the cylinders. Instances are known where so much lubricating oil reached the combustion space that when the engine was overheated it ran without fuel oil, merely by burning the lubricant. This shows why proper precautions should be taken to get as little oil on the cylinder walls as possible. An excess can do no good except act as a very expensive fuel, and is sure to gum up the piston rings. Splash guards are often provided to prevent an excessive amount of lubricating oil from finding its way to the cylinder walls.

The housings of cross-head type engines are closed by fitting sheet steel or cast iron plates between the frames, these plates being drawn down on cork or canvas gaskets to prevent leakage around the edges.

An objection frequently raised against the tightly enclosed housing is the impossibility of observing and feeling the bearings. In attempting to meet this objection, some builders resort to various devices such as electric flow indicators placed in the branch pipes to the main bearings; thermometers placed in the main bearing boxes, with temperature-indicating scale

at the operating platform; circumferential grooves in the ends of the main bearings to catch the overflow and conduct it through pipes to a point near a door in each splash plate, through which the temperature of the oil flowing out of the pipes can be felt with the hand. Each of these devices has the defect that it does not give an indication of the condition of the crank pin and wrist pin bearings, which are the bearings which are the most apt to heat.

Regulating the Distribution of Oil to Main Bearings. Unequal distribution of oil to the main bearings may be corrected by placing some sort of restriction in each of the pipes leading to the main bearings, such as a washer or plug with a small hole drilled in it, thus permitting only as much oil to flow into each pipe as will pass through the restriction. In addition to this the oil discharge line from the circulating pump should connect to the oil main at the center of its length rather than at its end. This arrangement insures the group of bearings under each cylinder getting the same amount of oil but the equal distribution of oil to each of the bearings in each group depends upon the correct proportioning of the oil passages and the maintenance of correct clearances in the bearings.

Main Bearing Clearances. The clearance of main bearings is particularly important. They will run noiselessly with considerable clearance and, without giving any indication of their looseness, they may allow so much oil to flow out of the ends that not enough will reach the crank pins and wrist pins. Little if any grooving of bearing surfaces should be done where forced lubrication is used; usually one longitudinal groove at the point of greatest pressure in each bearing being sufficient to distribute the oil.

The fitting of shims between the faces of the bearing shells should receive very careful attention. The edges of the shims abutting on the shaft or pin must be cut well back so that they will not interfere with the flow of oil around the bearing, but this relieving must not be carried out to the ends of the shims, otherwise large holes will be formed at the ends of the bearings and so much oil will flow out that the normal pressure can not be kept up in the system.

Check Valves in Connecting Rods. It may sometimes happen that a good flow of oil is obtained through all bearings when the oil circulating pump is operated while the engine is not in operation but the supply to the wrist pin bearing is insufficient when the engine is running. This may be due to the inertia effect of the oil in the connecting rod, whereby the oil is forced down out of the rod when the rod suddenly changes its direction of motion as it passes bottom center. This effect is most noticeable on high speed engines. On some American submarines this trouble was corrected by placing a simple ball check valve in the bottom of each connecting rod. This check valve promotes circulation of oil for the reason that when the rod moves upward the oil in the central passage is accelerated, the velocity thus built up assists the pump pressure after the rod passes top center and starts on the down stroke, and acceleration in the downward direction that would force the oil back against the pump pressure is prevented by the valve.

Cylinder Lubrication. The working cylinders are usually lubricated by means of a small high pressure mechanical lubricator which consists of a cast iron reservoir containing pumps or units which vary in number with the number of oil feeds desired, there being one unit for each feed.

Fig. 107 shows a cross section of the McCord lubricator through one of the pump units. Each unit consists of a cast body hung from the top of the reservoir, containing two pump plungers, "X" and "Y", connected at the top by a crosshead "Z", and actuated vertically by another crosshead attached to the stroke shaft "C", which in turn is driven by the eccentric "B". This eccentric is revolved by a ratchet mechanism located at one end of the interior of the reservoir and driven by a shaft extending through a stuffing box to the exterior.

In operation the primary plunger "X" on its upward stroke draws the oil from the reservoir through the intake and on its downward stroke delivers it to the sight feed "E", whence it is drawn by the delivery plunger "Y" on its upward stroke and forced to the point to be lubricated by the downward stroke of same. The amount of oil delivered by each plunger is entirely dependent on the length of stroke which is regulated

by the adjusting nuts "F", shown just above the driving yoke "Z". The lower these adjusting nuts are set the longer the stroke of the plungers, and consequently the greater the delivery of oil.

The oil delivered by these pumps is led to the cylinder walls through the jacket and water space by some form of fitting, a common design being shown in Fig. 108. Each cylinder is fitted with from two to four or more of these fittings which are equally spaced to distribute the oil evenly around the cylinder walls.

Centrifugal Banjo Lubricators. When gravity feed lubrication is used some form of lubricator such as the Banjo ring,

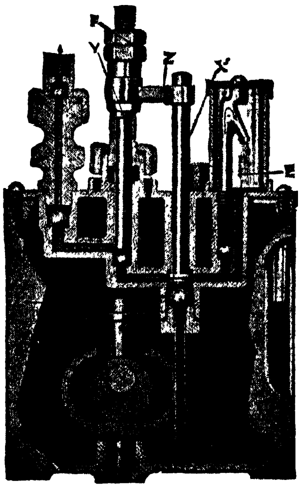


Fig. 107.—McCord Lubricator

Fig. 108 is used for carrying the lubricating oil to the crank journals. The oil is collected from around the main bearings, or else from a separate gravity fed supply, into rings, attached to the crank webs. By means of centrifugal force a positive flow of oil is made to the crank journals. When the lubrication of the wrist pin depends on this system check valves are always placed near the foot of the connecting rods.

Lubrication of Gears and Cams. A very heavy oil such as Crater compound is used for lubricating gears and cam rolls.

The heavy oil is applied with a brush to the cams and rolls. Gears that are exposed to water can be perfectly lubricated by Crater compound as this lubricant will not wash off or allow the water to come in contact with any metal upon which it has been applied.

Lubrication of Air Compressors. The same oil used in the cylinders and lubricating system of the Diesel engine should be used for lubrication of the air compressor cylinders, and the oil should be used in the smallest possible amounts. Trouble is always caused by the use of an excessive amount of oil and from the fact that the intake air contains too much

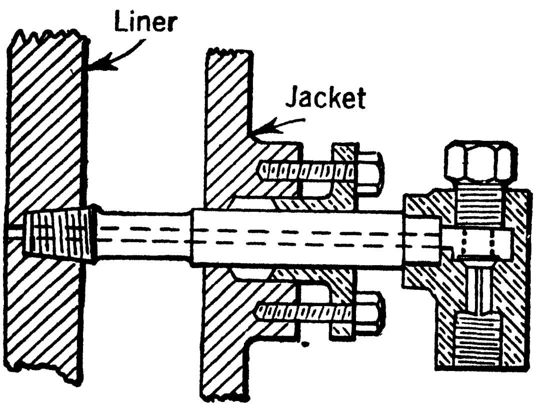


Fig. 108.—Cylinder Lubricating Fitting

moisture. If much water can be drained from the air bottles and if at the same time any difficulties are experienced with the cylinders or rings of the compressors, a mixture of five percent of whale or lard oil, which will cause the oil to emulsify and stick to the surfaces, will prevent any trouble.

Trouble from Water or Carbon in the Oil. By far the greater part of the lubrication difficulties with Diesel engines have been caused by water or carbon in the oil. For pistons above 23 inches in diameter some form of cooling must be used. Because of its superior heat absorption characteristic, water is the most desirable cooling medium, and because it is available in unlimited quantities, sea water is the logical choice for marine engines. The use of sea water, however, is re-

stricted to engines in which the designer has eliminated all possibility of water leakage into the crank pits, because even the best lubricating oils will emulsify with salt water and the emulsion thus formed will clog up the lubrication system. For this reason many designers have chosen fresh water for cooling the pistons, so that any small amount of leakage from the joints of the piston cooling system will do no serious damage.

Even with the fresh water system some trouble will be experienced, if the water gets into the oil, for an emulsion will in time be formed, though this emulsion can be broken down and the oil and water separated. In this connection it is interesting to note that this emulsion is very difficult to

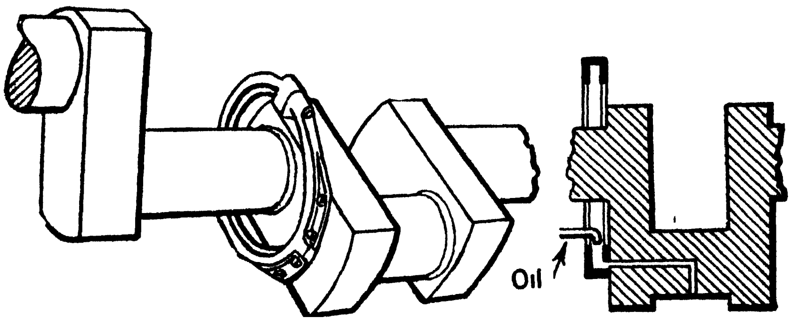


Fig. 109.—Centrifugal Banjo Lubricator

break down if it contains oxide of iron. For this reason all the metal parts inside the housing should be kept free from rust. This leads to some designers using a portion of the lubricating oil for piston cooling. With this arrangement lubrication troubles are sometimes caused by carbonized oil flaking off from the interior of the pistons and mingling with the oil in sufficient quantities to clog small passages in the lubrication system. The tendency to carbon formation on the underside of the piston top is greatly reduced, if the cooling oil is given a high velocity through the piston.

In trunk piston engines a great deal of carbon and so called "black oil" from the cylinders will work past the piston rings into the crank pits and mix with the lubricating oil.

This cylinder carbon interferes with lubrication both by retarding circulation in the system and by reducing the lubricating qualities of the oil. Although it is commonly referred to as carbon it actually consists of a mixture of carbon, metal dust and other impurities, such as sodium sulphide, that may be deposited out of insufficiently washed fuel oil.

Lubricating Oil Coolers. Some means of cooling the lubricating oil used in the lubricating and piston cooling systems must be used in order that a given quantity of oil can be used over and over again and that the oil used will be of a correct temperature to maintain an oil film of proper viscosity between the bearing surfaces.

Fig. 110 shows a typical oil cooler for cooling the lubricating and piston cooling oil. The cooler consists of a cast iron body having a water inlet and outlet header at either end. The inside cooling element consists of a large number of straight

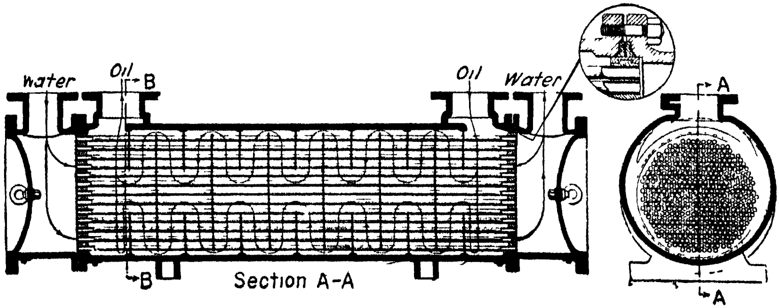


Fig. 110.—Lubricating Oil Cooler.

bronze tubes which are tinned on their outsides and are held in place by means of brass headers into which the tubes are expanded and sweated. The oil flows around the outside of the tubes in several passes and the cooling water passes through the tubes as shown by the arrows.

Circulating Water Systems. The function of the circulating water system is to maintain a maximum working temperature of the cylinders and the parts of the engine subjected to great heat. Theoretically it would be advisable to convert all the heat of the fuel into useful work. However, about one

third of the heat contained in the fuel is lost to the cooling water, in order that parts of the engine may not be injured by overheating and to make possible the efficient lubrication of the cylinders and main bearings.

Volume of Cooling Water Required. The volume of cooling water required varies with the size and type of engine. Four cycle engines require from 6 to 10 gallons per B.H.P. hour, while 2-cycle engines use somewhat more. The volume used depends on the initial and the terminal temperature of the water. The quantity varies in direct proportion to the load for any particular size of engine, and in all cases in inverse proportion to the difference in temperature between the water entering and leaving the jackets.

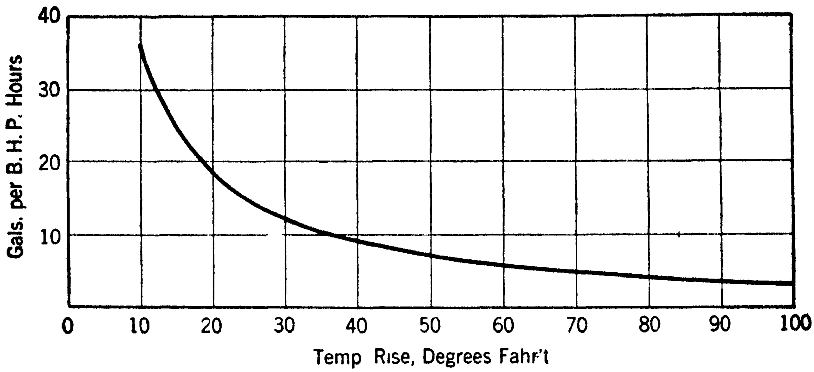


Fig. 111.—Average Water Required for Cooling System of 4-Cycle Diesel Engine

Between half and full load the jackets absorb about 2,000 B.T.U's per B.H.P. hour, or, on the basis of 60 degrees F. difference between the inlet and outlet temperatures. (For instance 70 degrees entering and 130 degrees leaving), the water circulated amounts to about four gallons per B.H.P. hour.

All this water is not lost as in the case of the non-condensing steam plant, or rendered unfit for further use as in the producer of a gas power plant, it may be recooled and re-used by means of a water cooling tower. Where a cooling tower is used the loss of water is very small, being represented by evaporation and spray blown from the tower.

Fig. 111 shows a chart of the average water required for

the cooling system of a 4-cycle stationary engine. It will be noted that the greater the temperature rise allowed the smaller is the quantity of cooling water required.

For cooling the compressor and engine a constant flow of water adjusted to the engine requirements, is desirable. This is best supplied from a constant head tank, such as in the system shown in Fig. 112, with the tank located at least 20 feet above the water inlet of the engine, so that the head of water will overcome any vapor tension formed within the engine jackets.

Path of Cooling Water. With most engines the cooling water is first passed through the air compressor coolers and jackets of the compressor, then through the engine cylinder jackets, and last through the cylinder head and exhaust manifolds. In other type engines independent connections are used, with branches from the main supply pipe to the compressor and the different working cylinders. Although the quantity of water required for cooling the cylinders is so much larger than that required by the compressor that it is not warmed ap-

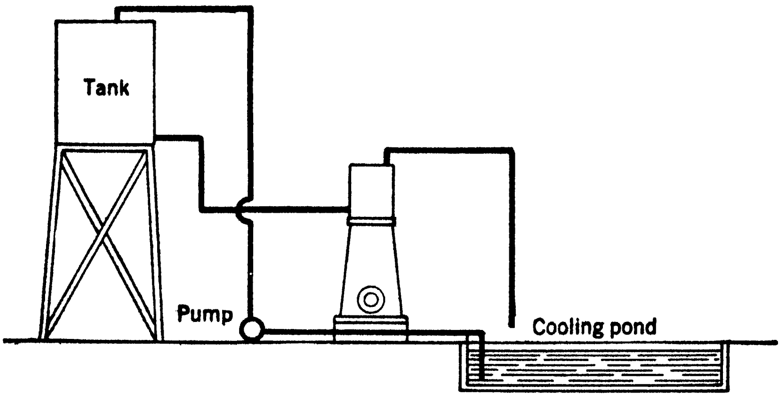


Fig. 112.—Cooling Water Tank

preciably by passing through the latter first, an independent water supply to the compressor has much to recommend it, affording easier adjustment of the supply of cooling water to the different cylinders and heads. Separate discharges for the water from the cylinders and from the cylinders head should

be provided, each with its own thermometer, so that the temperature of a cylinder or head can be controlled. The discharge pipes should be provided with valves, to regulate the discharge, as the water is under pressure.

In large installations, all the water discharge pipes are brought to a central discharge point and mounted on a board. A valve regulates the overflow, and a thermometer, mounted on the board above each respective discharge, indicates the temperature. A master valve controls the main supply to the engine. When the engine is shut down, only the master valve is closed, the other valve being left in adjustment.

Excessive Cooling Dangerous. Care should be used in adjusting the supply of cooling water to the engine needs, and an excess should be avoided, as it will cool the cylinders too much; they will contract, whereas the piston, being hot, will expand. Excessive cooling may thus cause piston seizures.

When the engine is stopped the water should be continued in circulation for some time, as the heat stored in the piston, the cylinders, and the cylinder head is considerable, being sufficient to bring the water to the boiling point, so that sudden stopping of the circulating system might cause the cracking of the cylinder or cylinder heads.

Cooling of Valves. Cooling around the exhaust valve is very important, since this part is subject to the intense heat of the exhaust flame. Exhaust valves of large size are very often separately cooled, the cooling medium flowing through a pipe in the valve stem to a pocket in the valve disc and returning through a channel around the inlet pipe, hose or other connections being used for the inlet and discharge of water to the valve stem.

Cleaning of Sediment from Cylinder Jackets. At the foundry cores are provided for the cooling water jackets of the cylinders and cylinder heads. The holes through which these cores are removed from the castings are sometimes plugged up and sometimes fitted with cover plates for inspection and cleaning. The cleaning necessary is due to salt deposits and scale. Scale forms most readily at the hottest places and is a thermal insulator which prevents the heat from getting to the cooling water. Trouble with overheated engines

and engines requiring an excessive amount of cooling water can often be traced to an accumulation of sediment and scale; especially is this so with marine engines.

Sea Water. When sea water, or other water containing lime or magnesia is used in the cooling water system, the terminal temperatures must necessarily be kept low—about 120 degrees at the maximum—in order to prevent the deposits of salt which will interfere with the efficient cooling of the cylinders and cause internal stresses to be set up which lead to cracked cylinders and heads.

Circulating Pumps. Pumps that are driven directly from the engine by beams, chain or gearing are usually of the plunger type. Independently driven pumps are almost always of the centrifugal type. Modern marine and stationary installations are usually equipped with independently electric driven circulating pumps. With these units the cooling water can be very closely regulated to give the best results. Also, after shutting down the engine the circulating pump can continue in operation in order to cool the engine down. It is generally best to cut down on the cooling water gradually as the engine cools, so as to hold the water discharge temperature at about the same amount as it was when the engine was running, and then to gradually cool the engine down. With attached pumps this cannot be accomplished without the necessity of an independent auxiliary pump.

CHAPTER IX

Indicator Cards and Engine Testing

The Indicator—Handling and Care of Indicator—Indicator Hook-ups—Indicator Cards—Interpreting Indicator Cards—Obtaining the Indicated Horse Power—Formulas—Electrical Horse Power—Brake Horse Power—Typical Indicator Cards—Compression Cards and Methods of Taking—Adjusting Compression Pressures—Engine Testing-Dynamometers—Fuel Consumption Tests, etc.

Indicators. The indicator, Fig. 113 is an instrument used to obtain a pressure-volume diagram of the gases in the cylinder of an engine, and from this diagram is obtained the mean effective pressure from which the horse power of the engine can be figured. Also, certain defects in valve settings can be seen and remedied.

The piston of the indicator is of carefully determined area and is accurately fitted to its cylinder so that it will move up and down without sensible friction. The cylinder is open at the bottom and fitted so that it can be secured to the indicator cock which leads into the engine cylinder, and by which arrangement the underside of the piston is subjected to the varying pressure of the gases acting therein. The upward movement of the piston, due to the pressure of the gases, is resisted by a spring of known resilience. A piston rod projects upward through the cylinder cap and moves a lever having at its free end a pencil point, whose vertical movement bears a constant ratio to that of the piston. A drum of cylindrical form and covered with paper is attached to the cylinder of the indicator in such a manner that the pencil point may be brought in contact with its surface, and thus record any movement of either pencil or paper. The drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord attached to the indicator gear and in the opposite direction by a spring within itself.

When this mechanism is properly adjusted and free communication is opened with the cylinder of an engine in operation, it is evident that the pencil will be moved vertically by the varying pressure under the piston, and as the drum is rotated by the reciprocating motion of the engine, if the pencil is held in contact with the moving paper during two revolutions of the engine, a figure or diagram will be traced, representing the pressure in the cylinder during the suction, compression, expansion and exhaust strokes of the piston.

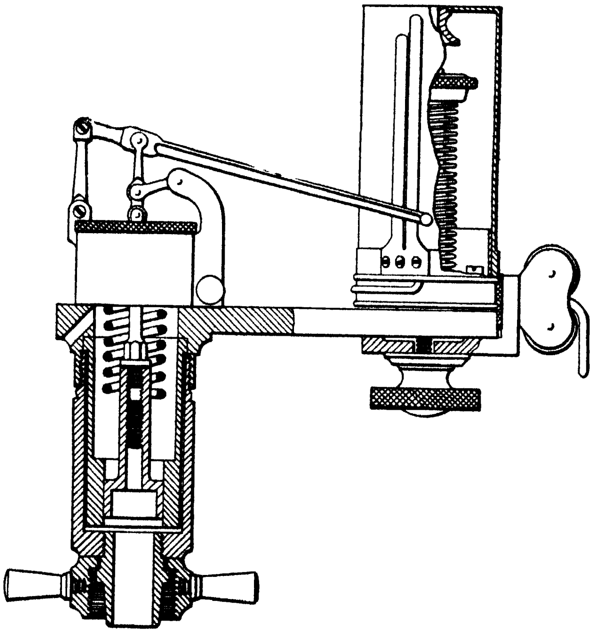


Fig. 113.—Crosby Indicator

Indicators for Diesel engines are usually provided with pistons having an area of $\frac{1}{8}$ square inch and with 600 pound springs calibrated for this size piston. With a compression of 450 pounds per square inch this would give a card having a height of $\frac{3}{4}$ inch.

The indicator is a very delicate instrument and in order to secure good results from its use it must be handled with care and kept in good condition. The instrument should never be left carelessly around, but should be placed in its case after

thorough cleaning. In order to get good results the piston should be removed and cleaned and oiled after taking one or two cards. Only the best grade of fine sperm oil should be used and all parts should be kept well oiled.

Under no circumstances should the indicator be allowed to operate for more than one or two cycles as its life is only good for a few minutes in actual operation. The clearance between the cylinder and piston of a Crosby indicator is only 1/20000 of an inch.

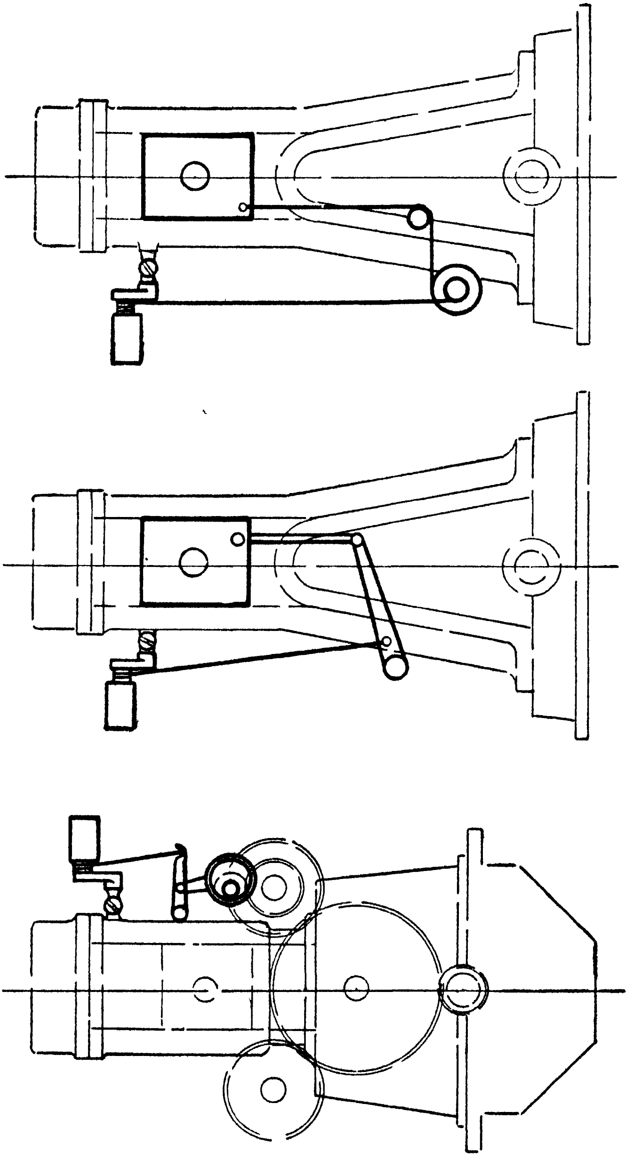
Indicator Drum Motion. The motion of the paper drum may be derived from any part of the engine which has a movement coincident with the piston. In steam practice the motion is derived from the cross-head as being the most reliable and convenient but with Diesel engines the majority use trunk pistons and so have no cross-heads. The movement of the engine piston, whatever it actually is, must be reduced to the length of the diagram, about 2½ inches. This reduced motion must be in exact ratio to the motion of the piston. To obtain this motion one of various devices are employed.

Fig. 114 shows the arrangement used on Nelseco engines. As this indicator gear is only engaged when taking cards, it is provided with a single position clutch on the cam shaft which is engaged when taking cards.

The arrangement shown in Fig. 115 has the advantage of accuracy of movement, simplicity of construction, and little vibration of cord as the indicator cord moves in the same direction as the piston.

Fig. 116 shows another way in which the reducing motion is obtained. The large wheel on which the cord from the piston works is fitted with different size bushings, or rings, by which the diameter may be increased or reduced in order to change the length of the diagram. There is a coil spring contained within the spring drum of the reducing wheel which keeps a tension on the cord from the piston at all times, and also keeps a tension on the cord to the indicator, from the reducing wheel. This type of drive can be fastened to the cross head also if the engine is of that design.

Diagrams should have full information marked on them such as date, time, cylinder, spring, engine number, load, r.p.m.



Figs. 114, 115, 116.—Indicator Driving Arrangements

and blast pressure, as well as any other data concerning adjustments if diagrams are being taken for this purpose.

A double atmospheric line, when a strong spring is in use, shows slackness in the indicator joints or an improperly secured spring. Irregularity in the compression curve may be caused by: an indirect drive over a stiff working pulley, the indicator cord rubbing on a pipe on edge of a casting, slackness in the indicator gear, incorrect length of indicator cord or by an error in the indicator. This latter, however, can be checked by removing the spring and ascertaining that the vertical line drawn by the pencil is in line with the axis of the drum and at right angles to the line made by rotating the drum with the pencil held stationary.

From a practical point of view the real value of the diagrams to the engine user is, to assist him in diagnosing the cause of any irregularity in the operation of the engine and of ascertaining the correctness of compression pressure, maximum pressure and equal development of power by each cylinder of a multicylinder engine.

It must be admitted, however, that diagrams can be affected so much by various sources of error that it is almost futile to ascribe an accuracy to them which they do not possess. When the diagrams are worked out, the manipulation of the planimeter also leads to further discrepancies, so that it is almost useless to state the M.E.P. or I.H.P. to a finer figure than is represented by 1%. This view means that figures based on indicated horse power such as the mechanical efficiency of the engine or fuel consumption per I.H.P. can only be considered as approximations, and consequently should not be taken too seriously.

To those not accustomed to take indicator diagrams the following hints may be of service. The indicator should be connected to the cock in such a position that the driving cord is as nearly as possible vertically over the driving pin, and further, the cord should run direct to the indicator drum. The small guide pulleys fitted on the indicator should not be used unless their use is unavoidable, on account of the friction and inaccuracies they are liable to introduce. If the foregoing conditions cannot be met, a light guide pulley which should

be reasonably large, say approximately two inches in diameter, may be used without danger of introducing serious errors, if placed correctly and free from friction.

The indicator cord must be flexible and should be thoroughly stretched before use, and should be so adjusted for length that there is no chance of the diagram being falsified at either end by the drum coming up against the stops.

The indicator spring should be of sufficient strength to limit the movement of the pencil to $1\frac{1}{4}''$ or $1\frac{1}{2}''$.

The indicator should be kept cool by removal from the engine when not in use. That is to say, it should be removed from the cock immediately the diagrams have been taken. The use of one indicator on a multicylinder engine tends to make this possible, and has the advantage, that the diagrams are more strictly comparable, as differences, due to the indicators are obviated.

The pressure of the pencil on the paper should be kept as light as possible consistent with the obtaining of clear diagrams when the pencil point is sharp. Accurate diagrams cannot be obtained unless this condition is observed.

For ordinary purposes it is preferable to have three or four super-imposed diagrams to show up possible irregularities, but for special investigation, a single diagram is always the best.

The indicator and its diagram, however, can be looked upon very much in the same light as the doctor views the human pulse, in that it gives an indication as to how the heart of the engine is working, and it can be considered of great value in getting all the cylinders to give an equal amount of work after a certain amount of overhaul has been given to the engine, such as removing the piston or taking down the fuel pumps.

Fig. 117 shows a theoretical, or ideal, indicator card of a four cycle Diesel engine, a practical card from the same engine and a diagram of the valve openings and closings and their relation to both cards.

The theoretical indicator card is obtained for any particular engine by the use of one general formula, which has as its basis the known action of gases under compression or expansion. By means of this formula, it is possible to compute a number of values representing the pressures in the cylinder

at different points in the cycle, and having obtained these values, to plot them or draw the curves of compression and expansion.

The practical card is obtained by means of the indicator as explained, and the ratio between these two cards, the theoretical and the practical, is known as the "card factor."

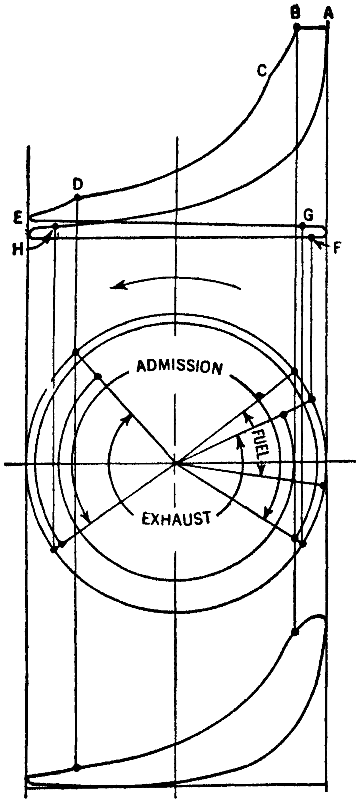
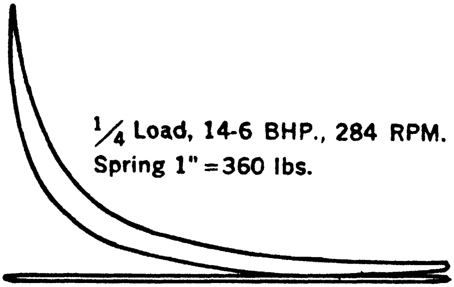
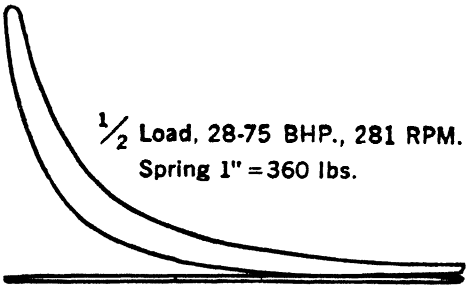
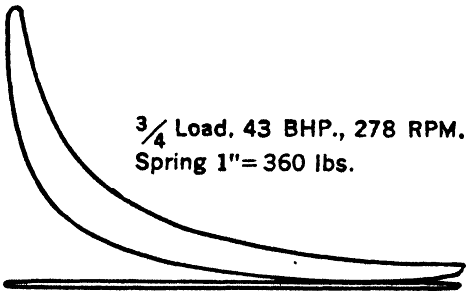


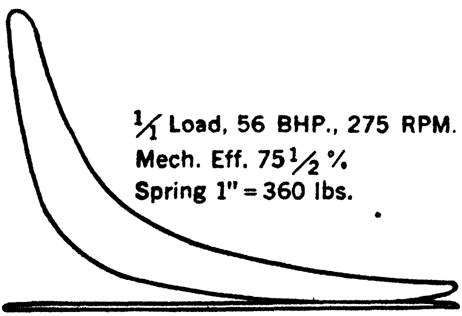
Fig. 117.—Theoretical and Actual Indicator Cards and Their Relation to the Valve Timing of a 4-Cycle Diesel Engine

The valve events shown are as follows: Fuel injection and initial combustion takes place from A to B; supplementary combustion from B to C; expansion of the gases from C to D; exhaust from D to E to F; air admission from G to H; and compression of pure air from H to A.

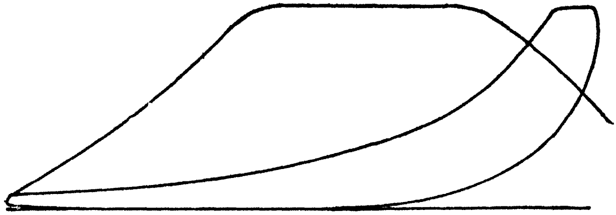




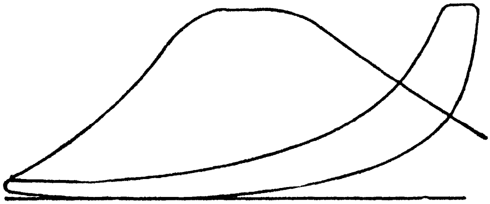




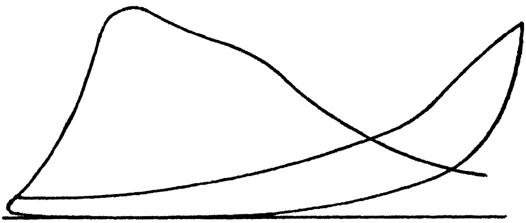
Figs. 118, 119, 120, 121.—Normal Indicator Diagrams



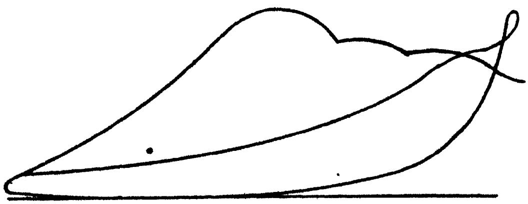
Overload Diagram.



Normal Full Load Diagram.

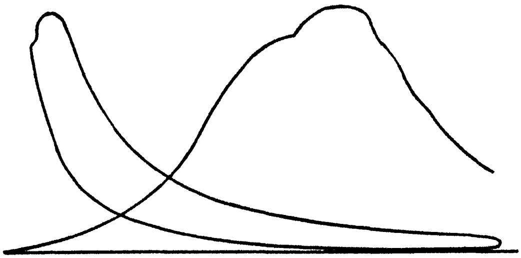


Combustion too slow. Blast Pressure too low or Lift of Fuel Needle too small or too much Resistance in Pulveriser.

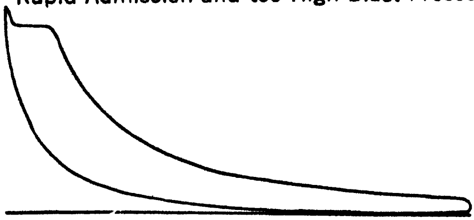


Fuel Admission too late Fuel Cam should be advanced.

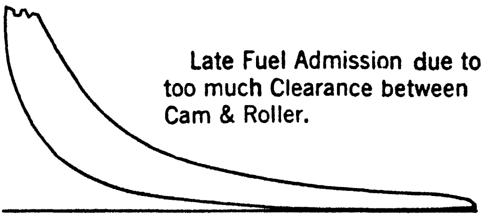
Figs. 122, 123, 124, 125.—Indicator Diagrams Showing Various Faults



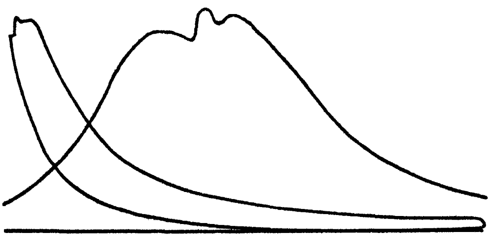
Abnormal Increase in Pressure due to too Rapid Admission and too High Blast Pressure.



Bulk of Fuel Late in Admission Due to Needle Valve Being "Shrouded" Thro' Being Ground in so much as to become "Bedded" into Fuel Valve Body.

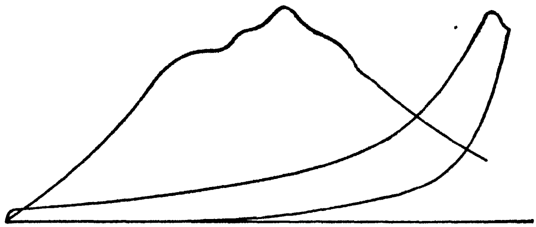


Late Fuel Admission due to too much Clearance between Cam & Roller.

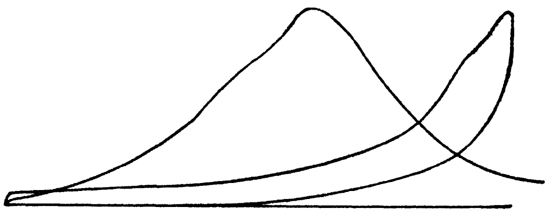


Late Fuel Admission with high Blast Pressure. Fuel Cam Should be Advanced and Blast Lowered.

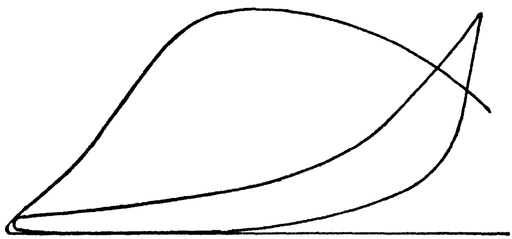
Figs. 126, 127, 128, 129.—Indicator Diagrams Showing Various Faults



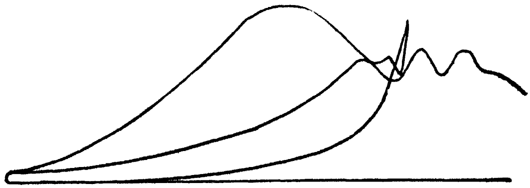
Explosions caused by too Little Resistance in Fuel Valve or using Bad Fuel



Combustion too Rapid caused by too high Blast Pressure or Lift of Fuel Valve too great



Early Fuel Admission due to Fuel Valve Being Set too Early.



Late Fuel Admission with Explosions. Caused by Fuel Valve Needing Adjustment.

Figs. 130, 131, 132, 133.—Indicator Diagrams Showing Various Faults

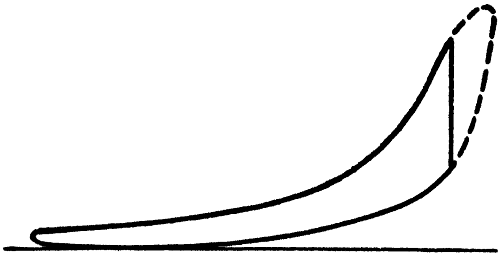


Fig. 134.—Too Long an Indicator Cord

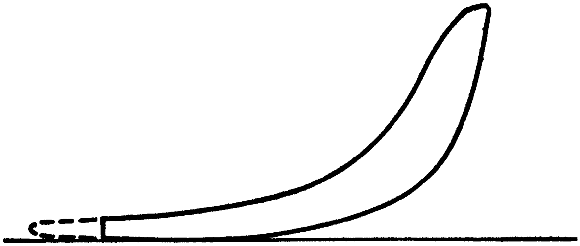


Fig. 135.—Too Short an Indicator Cord

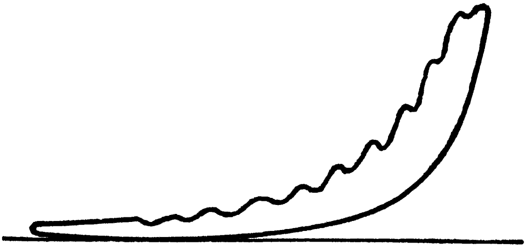


Fig. 136.—Excessive Vibration of Cord or Indicator Works Stiffly

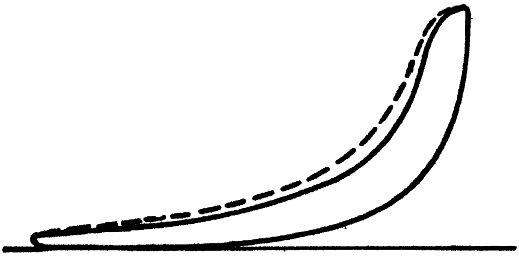


Fig. 137.—Leaky Valves

Interpreting Indicator Cards. The value of any indicator card naturally depends on the accuracy with which the pressure volume changes within the engine cylinder are recorded by the indicator. Good clear indicator cards often indicate at once defects or errors in the valve gear and valve setting, conditions in the spray valve, etc. so that the performance of the engine concerned can often be greatly improved by adjustments, simple in themselves, but the necessity for which it is difficult or impossible to detect by any other means.

To understand the action of the engine in relation to an indicator card and to enable the reader to compare good and bad diagrams, a number of diagrams are shown. These have been taken at various times from various engines and therefore show differences in length which would not have occurred had they all been taken from one engine with the same indicator gear and the same motion. Figs. 118, 119, 120 and 121 give a good idea as to the appearance of normal diagrams for varying loads.

Figs. 122 to 125 show how various indicator mechanisms can change the appearance of an indicator card. These two cards also show what is sometimes known as "the burning diagram." This burning diagram is obtained by moving the indicator drum rapidly by hand at the end of the compression stroke. To get this calls for little knack in operating the drum but it is very helpful and instructive in that it shows on a large scale the point at which the fuel ignites and how the pressure varies.

A number of the other diagrams shown in these cards, Figs. 126 to 133, have the burning diagrams drawn on them so that the value of the latter may be readily recognized. Little need be said about the other diagrams as the legend pertaining to each card is marked thereon. These diagrams, however, have been grouped together in this manner so that a ready comparison one with the other can be made.

If the indicator cord is too long a card such as shown in Fig. 134 will be obtained. If the cord is too short, the toe will be cut off as shown in Fig. 135. With the indicator working stiffly or if there is excessive vibration of the cord, a card such as shown in Fig. 136 will be the result.

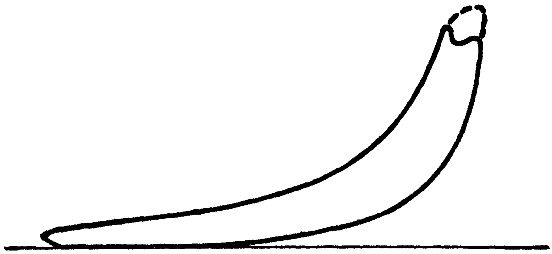


Fig. 138.—Too Rapid Injection of Fuel

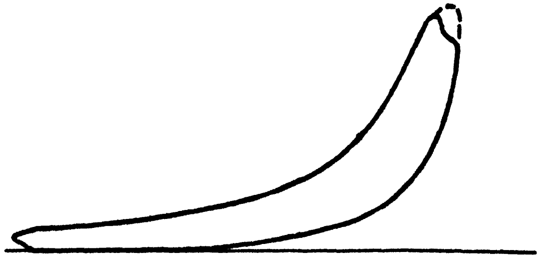


Fig. 139.—Late or Retarded Injection

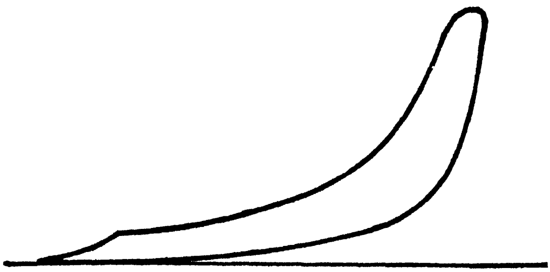


Fig. 140.—Early Exhaust

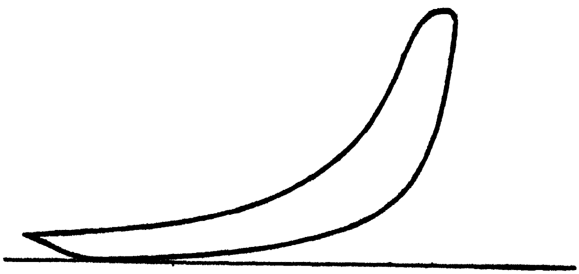


Fig. 141.—Late Exhaust

The diagram shown in Fig. 137 gives evidence of leaky valves or leaky piston rings, although such a card might result from a stiff working indicator.

Fig. 138 shows too early injection of the fuel, or the fuel is injected too rapidly. This may be caused by an insufficient number of atomizer plates or too large a hole in the atomizer button of the spray valve. Late, or retarded fuel injection is shown in Fig. 139. In this case full compression is attained after which, as the piston begins its return stroke, the expansion curve will follow the compression curve until fuel injection takes place. As can be seen, a large percentage of the card area is lost, depending of course, on the time of injection, and if the injection is greatly delayed, the re-expansion may reduce the temperature of the air charge to below the ignition point of the fuel, in which case there will be a misfire and on the following compression stroke a live charge will be in the cylinder, causing pre-ignition and explosion.

The effect of early exhaust is shown in Fig. 140 and that of late exhaust in Fig. 141. These cards are of no particular value as such should never be allowed to occur. Before taking indicator cards all valve settings and cam roller clearances should be thoroughly checked over.

Obtaining the Indicated Horse Power. After taking a set of indicator cards the mean indicated pressure is found by means of a planimeter, or by the average ordinates method.

This last method is used when a planimeter is not available and when this is done it is best to use a larger piston and cylinder, or a lighter spring, in the indicator, in order to obtain a card of greater area. The final results will be much more accurate when this is done.

Perpendiculars are erected at each extremity of the card from the atmospheric line. The distance between these two lines is divided off into a number of uneven spaces as shown in Fig. 142. Between each of these divisions a number of even divisions are made and lines are drawn parallel with the two perpendiculars, passing through the compression and expansion curves of the diagram. The distance between these two curves along each line is measured and their sum added and divided by the number of measurements taken, thus giving the

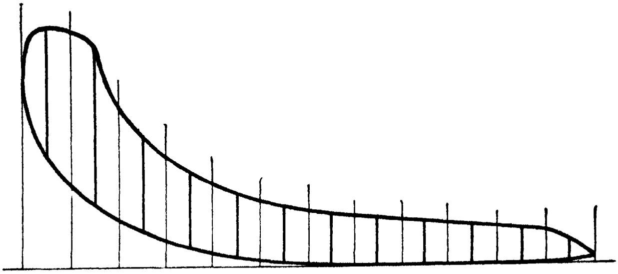


Fig. 142.—Obtaining the Mean Effective Pressure of an Indicator Card

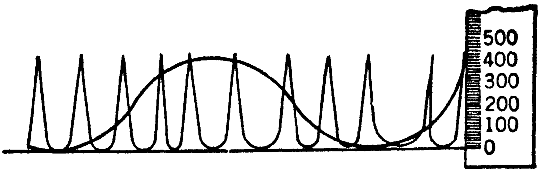


Fig. 143.—Compression Card

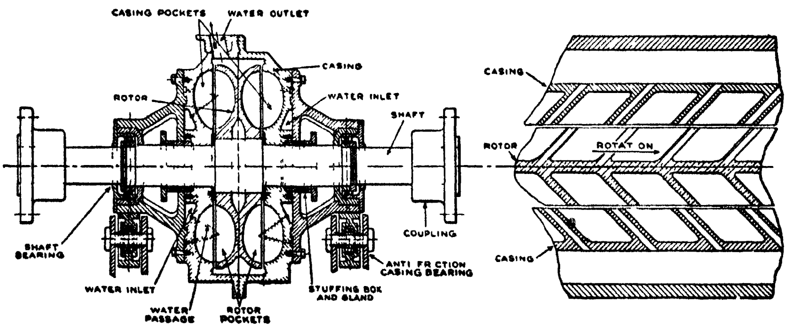


Fig. 144.—Cross Section of a Froude Hydraulic Absorption Dynamometer

average height of the card, or the mean indicated pressure of the card. The measurements are made by a scale graduated to read in pounds and to correspond to the indicator spring used.

To obtain the indicated horse power of each cylinder, the following simple formula is used,

$$\text{I.H.P.} \quad \frac{\text{PLAN}}{33000}$$

Where,

P = the mean indicated pressure per square inch.
L = Length of stroke in feet.
A = Area of piston in square inches.
N = Number of working strokes per minute.

As the mechanical losses due to friction and the power lost in operating the valves, etc. amounts to about 15 per cent, and the spray air compressor absorbs another 10 per cent of the power, it will be seen that the actual, or brake horse power, is considerably less than the indicated horse power.

Obtaining Electrical Horse Power of Engines. With Diesel driven generators, and aboard submarines, an accurate check can be made on the horse power of the engines at any given speed by means of the electric generators. The mechanical and electrical efficiency of the generators remains almost constant and, knowing this, the horse power can be found as follows:

$$\text{H.P.} = \frac{\text{Volts} \times \text{Amperes}}{746} \times \text{Generator Efficiency.}$$

Cylinder Constants. Much time can be saved in calculating the indicated horse power of an engine by the use of cylinder constants. As the working cylinders of any one Diesel engine will all have the same bore and stroke the constant will apply to each cylinder, at the same revolutions; only the mean effective pressure as figured from the indicator card will vary.

The constant for 4-cycle engines is found as follows:

$$\frac{L \times A}{33000 \times 2} = \text{Constant.}$$

For 2-cycle engines:

$$\frac{L \times A}{33000} = \text{Constant.}$$

Where $A =$ area of piston in square inches.
$L =$ length of stroke in feet.

Then, I.H.P. = Constant $\times$ revolutions $\times$ mean indicated pressure.

Example: Find the cylinder constant for the following 4-cycle engine.

Bore 12 inches.
Stroke 18 inches.

$$\text{Constant} = \frac{12^2 \times .7854 \times 1.5}{33000 \times 2} = .0257$$

Then, I.H.P. = .0257 $\times$ R.P.M. $\times$ M.I.P.

Cylinder Compression Cards. Compression cards are taken in order to find the compression in pounds per square inch in the working cylinders.

When taking compression cards it is good practice to let the engine run for a half hour or so to become thoroughly warmed up. The indicator is then attached to the cylinder and the fuel and spray air to that particular cylinder shut off. By opening the indicator cock and placing the pencil of the indicator against the card, and pulling slowly on the cord by hand, a series of peaks is obtained such as shown in Fig. 143. This is known as a "hot" card and is the proper kind to take.

A "cold" card is taken in the same manner, but the engine is turned over by means of starting air or, with submarine engines, by the electric motors. A hot card will give about 5 per cent greater compression than a cold card. A "fuel" card, or one taken with the cylinder firing on fuel, will give about 10 per cent more pressure than a hot card.

To read the compression a scale is used which corresponds to the size spring and piston used in the indicator. It is simply placed on the card and the pounds pressure read off from the scale.

Adjusting Compression Pressures. Proper adjustment of compression pressure values is important in order to have the compression equal to that desired, and also that all cylinders may have the same compression in order to equally balance the engine and divide the amount of work done in the cylinders,

thus insuring smooth running. Usually, compression values may be adjusted by inserting or removing shims from beneath the foot of the connecting rod and the crank brass. Engine builders issue tables showing the effects upon compression of various size liners. Such a table is shown below.

Compression pounds per square inch	Thickness to drop pressure fifty pounds	Drop in pressure due to .010 inch thickness of liner
700	.0375	14.
650	.0425	11.7
600	.050	10.
550	.060	8.3
500	.070	7.1
450	.082	6.05
400	.092	5.4

Example:

If pressure is 490 pounds, and you wish to make it 450 pounds: 490 — 450 = 40 pounds.

At 500 pounds, .010″ liner changes pressure 7 pounds, then liner required equals $\frac{40}{7}$ times .010 inch or .057 inch.

Engine Testing. The only accurate way to obtain the actual horse power developed by an engine is by means of the Prony brake or the hydraulic absorption dynamometer. These are shop tests and are carried out by the builders during the trials and tests of the engine at the place of manufacture.

Dynamometers. A cross section of a Froude hydraulic absorption dynamometer is shown in Fig. 144 and set up for test in Fig. 145.

The dynamometer is arranged for direct coupling to the engine to be tested and during operation the two shafts operate at the same speed. The dynamometer shaft carries a rotor in practically the same manner that a centrifugal pump shaft carries its impeller, and this rotor is free to revolve within the dynamometer casing, which casing is also free to revolve since it is supported on a set of bearings. Water is circulated through the casing and both the external face of the rotor and the internal face of the casing contain pockets separated by vanes. With the revolution of the rotor, therefore, the casing simultaneously tends to revolve, due to

the rotative force transmitted from the rotor to the casing through the circulating water. The rotation of the casing is, however, prevented by the suspension of the necessary weight from one end of a horizontal lever arm, while its other end is fixed to the casing. The circulating water thus provides an hydraulic resistance to the revolution of the rotor, and this resistance is increased to the maximum by means of the arrangement of the pockets and vanes on the opposing faces of

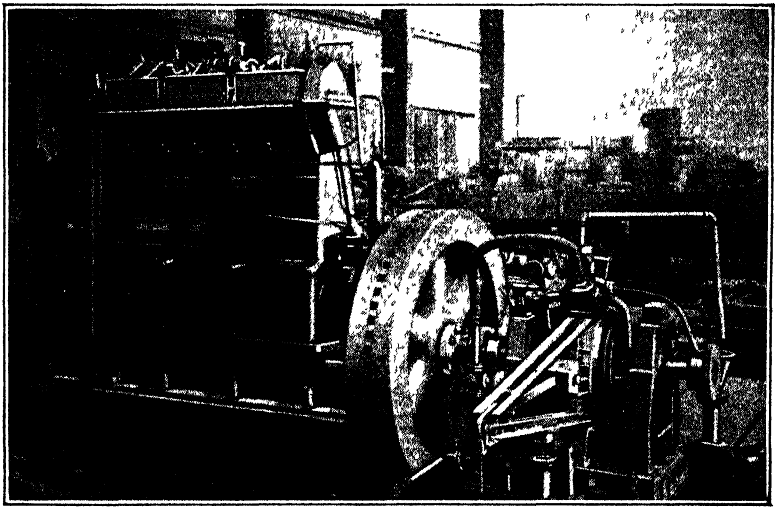


Fig. 145.—Froude Dynamometer Set Up for Test

the rotor and casing. In addition to furnishing this resistance the circulating water also provides a medium wherein the heat, developed by the absorption and destruction of the power transmitted to the dynamometer, is removed. The weight carried on the outer end of the lever arm is variable, and, for each operating condition, is made exactly such as will cause the casing to be in equilibrium or balance between its tendency to revolve in one direction with the rotor and in the opposite direction due to the downward pull of the weight on the horizontal lever arm. The entire machine is so constructed that the total force of rotation, transmitted by the engine under test to the dynamometer shaft, is in turn transmitted

without deduction to the dynamometer casing, and therefore this force becomes directly measurable with absolute accuracy. With the engine under test in operation and coupled to the dynamometer, and with the dynamometer casing in balance or equilibrium due to the weight attached to the outer end of the lever arm, the power developed by the engine is entirely absorbed by the dynamometer and is directly proportional to the weight.

The power absorbed by the dynamometer for any given testing operation is computed by means of the following formula:

$$B.H.P. = \frac{W \times N}{K}$$

In this formula, W is the statically unbalanced weight in pounds of the removable weights which have been suspended from the long lever arm in order to balance the tendency of the casing to rotate due to the operation of the dynamometer; N is the operating speed of the engine under test, and also the dynamometer shaft and rotor, in revolutions per minute; and K is a constant, for any given dynamometer, which depends on the exact length in feet between the center line of the dynamometer shaft and the center of gravity of the removable balancing weights attached to the lever.

Fuel Consumption Tests. The practical operating engineer is only called upon to make such simple tests as to demonstrate the fuel and lubricating oil consumption per electrical horse power hour, in the case with stationary engines driving generators, and the speed in knots at different revolutions, with marine engines.

When a test is to be made the engine should be thoroughly examined and note made of the general condition of the valves and bearing surfaces, valve timings, fuel pump settings, cam roller clearances, etc.

Compression cards should be first taken and all cylinder compressions evened up to the builders specifications. All instruments and apparatus used during a test should be checked up and their accuracy verified by test or comparison with a recognized standard.

The duration of a test will depend upon the character and

objects in view, and in any case the test should be continued until the consecutive readings of the rates at which the fuel is consumed, taken at say half hourly intervals, become uniform and thus verify each other.

The engine should first be run a sufficient time to bring all the conditions to a normal and constant state. Then the regular observations of the test should begin, and continue for the allotted time.

During the test the engine should be kept at a certain constant speed and load. Indicator cards should be taken at intervals of fifteen minutes and should be taken simultaneously on all cylinders if possible. All data obtainable should be taken such as pressures of the different stages of the spray air compressor, pressures of the circulating water and lubricating oil systems, temperatures of circulating water injection and outlet and lubricating oil discharge from piston cooling system if so equipped, temperatures of the exhaust gases and the color of same.

An accurate account is taken of the fuel oil consumed and this can best be taken from a tank which is filled to its original level at the end of the test, the amount of fuel required to do so being weighed.

CHAPTER X

Operation of Diesel Engines

Upkeep of the Engine—Starting and Running Instructions—Preparations for Starting—Regulating Fuel and Spray Air—Critical Speeds—Securing the Engine—Failures in Ignition—Mis-firing in Cylinders—Smoky Exhaust—Engine Knocks—Cylinder Explosions—Scavenger Receiver Explosions—Crankcase Explosions—Excessive Temperature Rises — Lubricating System Troubles — Circulating Water System Troubles—Air Compressor Troubles.

A Diesel engine, when in perfect condition, has very few troubles. If it were possible to give it perfect attendance, an engine would come near being in perfect condition, and, therefore, give perfect performance. Such, however, is only an ideal which can be approached more or less closely, depending upon the amount of care and attention bestowed upon the engine. A thorough understanding of the principles of operation of an engine is the first requisite; afterwards, constant attention to every detail, however small it may appear, on the part of the engineer in charge. This is essential to obtain the best results.

The Chief Engineer should keep a "log" book and make note of every minor defect and adjustment of his engine. A regular routine of cleaning and overhauling should be established and followed out as closely as possible. Absolute cleanliness of an engine is essential, as with any other type of high grade machine. It is impossible to expect results unless all parts are clean and adjusted in accordance with the way they were designed.

It is poor policy to let an engine run until it is absolutely necessary to give it a complete overhauling. It is better to lay out a systemmatic routine of overhaul, taking one cylinder in turn, and overhaul all valves, piston and bearings and make a thorough examination and adjustment of all the parts.

Starting and Running Instructions. Owing to the difference in construction and operation of different types of Diesel engines it is impossible to give operating instructions that will apply to all types of engines. The following directions, however, will apply to the average small 4-cycle, mechanically operated air starting, Diesel engine.

Preparations for Starting. Examine carefully all the spray, inlet and exhaust cams and rollers to see that they have the proper clearance and work freely.

Examine the controlling gear to see that it is working properly, and that nothing is sticking or binding.

Examine the air starting valve gear to see if it works freely and has the proper clearance.

Make a general examination of the rest of the engine, and be sure that there are no loose nuts or bolts, and that there are no tools lying around where they might jar into the working parts of the engine.

See that all shut off valves on spray air header to spray valves are open, also that all spray valves work freely in their stuffing boxes, and lubricate them by hand. If the engine has been standing idle for any length of time, or has received no attention after a prolonged run, care should be taken to see that the interior of the spray valves are not obstructed with a deposit of carbon.

All journals and joints of the fuel measuring pump gear and all valve gear should be lubricated by hand.

See that there is sufficient lubricating oil in the sump tank and, if the engine is installed on board ship or under conditions where salt water is used as the cooling medium, test the oil for the presence of salt water with nitrate of silver.

Pump up the fuel oil service tanks and after settling, drain off any water that may accumulate.

Open all suction and discharge valves for the lubricating and cooling system.

Open the indicator cocks and jack the engine around two complete revolutions. Close indicator cocks.

Open the fuel check by-pass valves and prime the fuel lines with the hand priming pump until a good stream of fuel is seen returning from each valve. Close the by-pass valves on

all cylinders not equipped for air starting. Make two or three strokes with the hand priming pump to force a small quantity of fuel into the spray valves.

Throw over the air starting lever to bring the air starting cams under their rocker arms. Jack engine over until one air starting valve has opened, about 15 degrees past top center.

Close the spray air header valve. Charge the spray air flask from the starting tanks to about 600 pounds pressure.

Turn on the starting air from the air flasks to the main air starting control valve, first making sure that this valve is closed.

Set the fuel control lever wide open. Slightly open the spray air control valve.

Care should be taken to see that no load is on the engine when attempting to start. In the case of the engine taking its load through the clutch, then the clutch lever should be placed in neutral position, or if the engine is driving a generator then the switches at the switchboard should be left open, and no load put upon the engine until it has at least reached its normal revolutions.

When the word is given to start, the main air starting valve is opened and the engine is started up by means of compressed air. If it is a four cylinder engine, two cylinders will be found to be equipped with air starting. A six cylinder engine will usually have three, and an eight cylinder engine four, cylinders so equipped, and in the case of directly reversible marine engines, all cylinders will be fitted with air starting valves to enable a start to be made from any position. As soon as the engine has turned over a few revolutions the cylinders will start to fire on fuel. The air starting lever is then thrown out and the fuel check by-passes closed on the air starting cylinders. All cylinders will then operate on fuel. The engine speed is then controlled by means of the fuel control lever and spray air control.

Observations of temperatures and pressures and R.P.M. should be logged. Any abnormal pressures or temperatures should lead at once to an investigation to locate the cause.

Soon after starting always feel the starting air supply pipes to the cylinders. If the starting valves leak, the hot

gases from the cylinder will back up into the supply pipes and heat them up. This can sometimes be remedied by turning the valves on their seats, but in case this does not stop the trouble the engine should be shut down in order to avoid damage to the valves, and the valves then made tight.

Occasionally feel the cylinder barrel just below the water jacket. Too high a temperature indicates hard rubbing between the piston and the cylinder or else a hot wrist pin.

Turn the spray valve needles on their seats occasionally to even up the wear on their seats. The inlet and exhaust valves should also be turned a quarter turn once each day.

See that the spray valve needles work freely at all times. If the valve stem shows a tendency to stick, a little kerosene poured on the stem will often free it, but if the valve does not respond readily to this treatment, the engine should be stopped and the valve made right. It is bad practice to set up on a spray valve stuffing box while the engine is in operation, as this results in sprung valve stems.

The lubrication should be carefully watched. An excessive quantity of lubricating oil is not required, especially in the air compressor cylinders. Oil the faces of cams with special extra heavy oil, applied with a brush.

Close watch should be kept on all air unions and joints to prevent waste of spray air.

If through some cause or other the cooling water should attain a temperature of 200 degrees or so, or boiling, large quantities of cold water should not be admitted suddenly, as this will cause the cylinders to cool suddenly and contract and grip or seize the pistons which cool more slowly. In case of overheating, the engine should be allowed to cool gradually.

All strainers should be regularly cleaned, including the fuel oil filter. The spray air separator should be blown down every hour or so.

In cold weather, after shutting down, the engine should be drained of all water to prevent freezing and bursting of the water jackets.

Immediately after shutting down, an engine should be cleaned and any small defects which were noted while in operation repaired.

Regulation of Fuel and Spray Air. Probably the most important thing about the handling of a Diesel engine is the regulation of the fuel and spray air. The air in the working cylinder is compressed, during the compression stroke, to about 450 pounds pressure per square inch. This high compression produces a temperature of about 1000 to 1100 degrees, Fahrenheit. Just about at top dead center, the spray valve is opened, and the spray air blows a fine spray of fuel into the cylinder. This fuel, as it meets the very hot air in the cylinder, is ignited and burns. If the fuel burns perfectly, it will leave no smoke or carbon. There may be a little blue smoke, caused by lubricating oil burning on the cylinder walls. Every part of the engine has to be working exactly right, or the fuel will only partly burn, with the result that there will be a smoking exhaust. This will foul up the cylinder heads, pistons, rings and valves with carbon deposit, and if allowed to continue will cause great loss of power. For this reason it is necessary to regulate the fuel and spray air pressure together, and sometimes even the cooling water, in order to arrive at the best combination for running at different loads and speeds. There is nothing gained in giving the engine more fuel than it will burn properly at a certain speed or load, as it is simply a waste of fuel.

If the engine is giving off black smoke from all cylinders, when there is a high spray air pressure and the compression in the cylinders is correct and the spray valves are correctly adjusted and functioning properly, the chances are that the engine is overloaded, or that it is getting too much fuel. In some cases, however, even when all parts are functioning properly, the engine will smoke considerably, in spite of everything that can be done, due to fuel of very poor quality.

At full speed and power, an engine should get all the fuel it will burn without smoking and the spray air should be about 900 to 1100 pounds pressure per square inch. If a lower pressure spray air is carried at full load, it will cause the engine to smoke, just as too much fuel does, because it will not blow the fuel into the cylinder fast enough to be properly burned.

If the supply of fuel is cut down, and with it the load and

speed, the spray air pressure must also be reduced. If the speed of the engine is reduced while pulling a heavy load, such as, in the case of marine engines running into a heavy gale or towing another ship, it may be possible to give the engine considerable fuel, but the spray air pressure must also be reduced. The reason for this is that the spray valves are timed to open a few degrees before top dead center in order that, at full speed, there will be time to get some of the fuel into the cylinder and actually burning as the piston goes over top center. This is analogous to the advance of the spark in a gas engine at high speed, or to the "lead" of a steam engine valve. In such cases where the speed drops and the full spray air pressure is kept on, this high spray air pressure will blow the fuel into the cylinder too quickly before top dead center, causing a knock or pound in the cylinder. If, on the other hand, the supply of fuel is cut down and the engine run at light load with high spray air pressure, there may not be sufficient fuel injected to produce the knock of pre-ignition but the powerful blast of spray air coming through the spray valve with the small amount of fuel will have a retarding effect upon the burning fuel oil, causing a white smoke and mis-firing in one or more cylinders if the load is very light.

The spray air pressure must be regulated to give the best results at different loads and speeds. Sometimes, due to some peculiarities of the engine or to certain spray valves, it is necessary to regulate the spray air shut off valves to each spray valve, in order to change the quantity of spray air to certain of the spray valves.

When the fuel is cut down it is also well to cut down the supply of cooling water to the cylinders so that they are kept at a good working temperature. It is a mistake to think that an engine runs best when kept the coolest; within certain limits they will run best when kept very warm. The proper regulation of the fuel control, the spray air control, and the temperature of the cooling water discharge are things which must be learned by experience.

Critical Speed. It will be found that there are certain speeds at which an engine runs best, and also other speeds at which it runs badly and with excessive vibration. This last is

called a critical speed, and should be avoided as much as possible as the vibration is dangerous to the engine. An engine should always be run at a faster or slower speed than the critical speed. On some engines the critical speed is marked on the tachometers.

Shutting Down the Engine. After a run, when the engines are to be finished with for the day, the cooling and lubricating oil pumps are run for a time to cool down the engine and engine room. It is best to cut down on the pumps gradually, if separately driven, and to keep the temperature of the cooling water discharge and lubricating oil return at about the same as running temperature. When the engine begins to cool down, the pumps may be slowed down and finally stopped.

The engines should be well wiped down, and any necessary repairs and adjustments made in order to be ready for the next run. In cold weather the cooling system should be drained to prevent freezing and bursting of the cylinder jacket and water piping.

The following are some of the more common troubles and their causes and applies to most all types of engines.

Ignition Does Not Take Place. When an engine is turned over by starting air and refuses to start up on fuel there may be several causes.

The fuel lines leading from the fuel measuring pump to the spray valves may not be primed or may have become "air bound". In this case, the fuel lines must be primed by means of the hand priming pump; first opening the fuel check valve by-pass and pumping with the hand priming pump until a solid stream of fuel shows through the return fuel sighting device. Be sure that the hand pump is then cut out and the fuel measuring pump properly cut in.

There may be air in the fuel lines or pump. Prime the fuel lines again. If bubbles show up from the by-passes every time the lines are primed it is a sure sign that the spray air is getting back into the fuel system. This means that the fuel check valves on the spray valves are leaky, stuck open, have dirt in them, or need grinding. Examine each one separately until the fault is discovered and correct accordingly.

The fuel pump valves may not be working properly and may be stuck or dirty. These should also be examined and the fault eliminated.

There may be water in the fuel lines. The water will have to be drained off from the engine service tanks and the fuel lines primed until the fuel containing water is worked out of the fuel lines.

The by-passes on the spray valves may have been left open or the spray air shut-off valves may be closed.

The fuel lines may be stopped up with asphalt or other impurities, and will have to be cleaned out before the engine will start.

The spray air pressure may be too low or shut off. When the spray air pressure is lower than the cylinder compression pressure, the hot air from the cylinder will enter the spray valve and cause combustion of fuel on the spray valve body, or blow unconsumed fuel into the air lines, which later may be ignited either from H. P. air from the compressor or from the actual leakage of the spray valve, as the combustion therein further spoils the valve seats, in which case, burst air pipes will almost certainly result. For this reason the spray air flask should be primed with at least 600 pounds of air before starting the engine.

When using heavy oil, in extremely cold weather, the oil may be so cold and viscuous that it will not flow freely, and will cause sluggish action in the fuel pump.

Mis-firing of the Cylinders. A cylinder mis-firing can be detected by an irregular sound of the engine, or by the jumping of the ammeter needle if the engine is driving a generator. To ascertain which cylinder is misfiring, the indicator cock, or cylinder pet cock, is opened, and if the cylinder is not firing properly a puff of smoke will escape. A clear, hot exhaust from the indicator cock shows that the cylinder is firing properly.

The fuel lines leading to a misfiring cylinder may be air bound, caused by leaky fuel check valves, or the fuel pump suction or discharge valves may be hung up.

The spray valve may be clogged up. In this case the spray valve must be removed and cleaned. The fuel check

valve or air check valve may be jammed shut. Sometimes the spray valve stem becomes broken and, while the valve stem appears to open and close properly, the lower part is held tightly to its seat and no fuel will reach the cylinder.

If a cylinder misfires when running at light load, or throttled down, it may be caused by a too close adjustment of the fuel measuring pump suction valve lifting mechanism. The screw which strikes this suction valve of the pump may be screwed out a little while the engine is in operation, to see if the missing cylinder will fire. If this is done, it must be carefully noticed that the same cylinder does not get too much fuel, and smoke, when running at full load.

Too much cooling water will cause a cylinder to misfire when running at light load. The discharge water from the cylinders should be kept at the proper temperature.

Misfiring is often caused by low cylinder compression. In order to properly ignite the fuel there must be good compression. At periodic intervals compression cards should be taken of all working cylinders and the necessary adjustments made by inserting or removing shims from between the foot of the connecting rod and the crank brass. With air injection engines compression pressures of less than 425 pounds will often cause misfiring.

Loss of compression is often caused by leaky intake and exhaust valves, and worn, broken or stuck piston rings.

Smoking Exhaust. A black smoky exhaust is often caused by low pressure spray air when running at near full load. Increasing the pressure is of course the remedy in this case.

If some cylinders are misfiring they will cause the other cylinders to smoke, due to the increased load placed upon them. Dirty spray valves, poor or dirty fuel oil, considerable carbon in the combustion chamber and exhaust manifold, broken or enlarged hole in the atomizer button, overloading etc. will all cause a smoky exhaust.

One cylinder may smoke due to overloading, or receiving more fuel than the others. This can best be ascertained by taking indicator cards of all cylinders and the necessary adjustments made to the fuel measuring pump to proportion equally the fuel to all cylinders.

Another cause of smoking is that of insufficient air in the cylinder to properly burn the fuel. This is usually caused by too great a roller clearance, causing the inlet valve to open late and close early. In the case of 2-cycle engines a low scavenging air pressure would cause the same effect.

White or bluish smoke is usually associated with too much lubricating oil supplied to the cylinders. Broken wiper rings at the bottom of the piston will allow an excess of oil, thrown up from the cranks, to work into the cylinder, or the rings at the bottom of the pistons may be gummed up, allowing oil from the cranks to work up into the cylinders. This is not likely to happen on all cylinders at once. The cylinders affected may be determined by opening the cylinder pet cocks and noting the exhaust.

An excessive spray air pressure at light loads will cause a white exhaust, similar to that caused by an excess of lubricating oil.

One cylinder that may be smoking more than the others may be getting too much fuel. Try screwing in the suction valve screw of the fuel measuring pump for that cylinder a very little, but make sure that the same cylinder does not go dead when the engine is throttled down.

Engine Knocking. If an engine knocks in all cylinders at every power stroke, it may be caused by an excessive spray air pressure for that particular load and speed. This is known as an "air knock".

An overloaded engine will cause a "fuel knock" on all cylinders, or in any individual cylinder that is overloaded.

Fuel rapidly injected into a cylinder will also cause a "fuel knock". This may be caused by an insufficient number of atomizer plates or broken or enlarged nozzle plate, allowing the fuel to be injected too rapidly and resulting in an explosive burning of the fuel.

If a spray valve sticks open, the fuel will be injected into the cylinder as soon as delivered to the spray valve by the fuel measuring pump. This causes pre-ignition and serious "knocks" and great strains are set up in various parts of the engine. Crankshaft failures are sometimes caused by preignition of the fuel.

The spray valve roller may not have enough clearance, causing it to either not close tight, or else to open too soon, giving a pre-ignition and "knock". Check up on the roller clearances of the spray valve, and also check the timing with the mark on the flywheel. The timing of the spray valve should be accurate to within one degree. The best way to tell the exact time of opening of the spray valve is to put an air pressure on the spray line and jack the engine over slowly by hand, and listen for the hissing of air. This will be the exact point of opening of the spray valve.

The holes in the atomizer of the spray valve may be worn too big, or the nozzle may have become worn to a larger diameter or become broken off. Anything that allows the fuel to enter the cylinder too quickly will cause an explosion and "knock" in the cylinder.

A misfiring cylinder will cause a "knock" in the other cylinders. All cylinders should be firing regularly before an attempt is made to locate knocks in an engine.

Carbon deposits in a cylinder and cylinder head, due to bad combustion or a smoky exhaust for some time, will cause "knocks" in that cylinder.

Too high a combustion in a cylinder will cause "knocks" and may be due to fitting new bearings or piston or connecting rod, without checking up on the cylinder compression and adjusting by means of the connecting rod shims.

If an engine knocks rapidly on all cylinders during all strokes of the pistons, the engine should be stopped at once, or very serious damage may result. In all probability there is something wrong with the lubricating oil system and the bearings have run dry and are beginning to seize.

When a bearing is beginning to seize, it may be revealed by local heating around that part of the housing where the knocking is heard. A piston beginning to seize may be heard or felt at the cylinder, just below the water jacket. In all such cases the engine should be stopped at once, and the trouble located. The part which has been running without oil must be oiled copiously and cooled down, and the fault in the lubricating system corrected.

A bearing which has become worn loose, or burnt out, will cause a "knock". This may be a main bearing, a crank bearing or a wrist pin bearing. The engine should be stopped at once and the various bearings examined.

If anything is hitting or interfering in the frame it will cause "knocks". Stop the engine and take the plates off from the frame in the vicinity of the knock, then jack the engine over slowly and look for some part hitting or interfering.

Too much roller clearance on the inlet or exhaust valves may cause a cylinder to "knock", because it does not get its inlet air quickly enough in the first case, and does not expell its exhaust gases quickly enough in the second case.

With some engines there has been trouble with lubricating oil getting up through the breather pipe from the crankcase into the intake header, and then into one or two cylinders nearest this pipe. This will cause a "knock" due to pre-ignition.

Cylinder Explosions. Explosions in the working cylinders, causing the relief valves to lift, may be caused by the spray valve packing being too tight and seizing the spray valve stem, or to a warped valve stem. When explosions occur the fuel check by-pass valves should be opened immediately, before further damage is done by the explosions. The spray valve stem should then be worked around with a wrench. If it can be made to operate it should be allowed to work for a few minutes before closing the fuel by-pass valve. If it continues to stick, the packing should be eased off a bit and kerosene squirted in around the valve stem to loosen it. Continued sticking may be caused by a bent or sprung valve stem. In this case a new stem must be fitted, as they are very difficult to straighten.

Occasionally also, the spray valve spring may be broken, allowing the valve to stay open, in which case cylinder explosions are bound to result.

It is not unusual for the relief valves to lift when first starting up an engine, due to the fuel being injected during the first few compression strokes, but not ignited. When the fuel is ignited it explodes and causes the relief valve to lift. This is especially true if the compression is low and the engine

is cold. It is not serious and will not occur after a few revolutions.

If the fuel check by-pass valves are not opened up when priming the fuel lines, preliminary to starting up, fuel will be pumped into the spray valve body in excessive quantities and, when injected into the cylinder, will cause explosions.

If a relief valve continues to lift, and the reason can not be discovered, the relief valve should be removed and tested to see if it is correctly set. The usual setting is for the valve to lift at about 750 pounds pressure per square inch. The practice of "setting up" on a relief valve to keep it from lifting, should be avoided as it is the only means of relieving excessive pressures in the cylinder and it is in every sense a safety valve, analogous to that of a steam boiler. Relief valves should be periodically set and tested every few months.

Scavenge Air Receiver Explosions. Explosions sometimes occur in the scavenge air receiver of 2-cycle engines. Scavenge valves may stick open when they have become choked up with carbon after continuous running with a black exhaust. This will allow the hot air of compression to enter the scavenge air receiver and if an amount of lubricating oil vapor is present, an explosion may take place. There is always a quantity of lubricating oil vapor, or fog, in the crankcase of Diesel engines, which, in the case of stepped piston engines, is often drawn into the scavenging cylinder. If there is excessive clearance between the scavenging piston and cylinder walls, and should a flame make its way from a working cylinder into the scavenging cylinder, an explosion may take place, resulting in more or less damage to the engine.

Crankcase Explosions. These explosions usually result from leaky piston rings; the flame passing down into the crankcase and igniting the oil fog or lubricating oil vapor.

The Scavenge Air Pressure Rises. With 2-cycle engines a rise is the scavenge air pressure is usually caused by the scavenge ports in the working cylinders becoming stopped up with carbon, producing a greater resistance to the flow of the scavenging air, and thus causing the pressure to rise. Another reason is that of an excessive back pressure on the exhaust line, such as might be caused by partly opened exhaust valve

or by the exhaust passages and muffler becoming filled up with carbon and soot.

In case of a loss of scavenge air pressure it may be caused by leaky rings, or relief cover plates may be opened slightly.

Exhaust Temperature Rises. The engine may be overloaded. There may be an insufficient supply of cooling water to the engine. Note the thermometers in the cooling water system.

Lubricating System Pressure Drops. The suction strainers or main strainers in the lubricating system may be plugged up. If, however, the strainers are on the discharge side of the pump this would cause a rise in pressure. The strainers should be cleaned regularly.

An insufficient supply of lubricating oil in the sump tank, would also cause a drop in pressure.

Some by-pass or cross connection valve may be open, and so cause a drop in pressure, or there may be broken or leaky pipe on either side of the pump. If it is on the discharge side, the leak can be easily seen from the oil leaking out; if on the suction side the leak can be detected by the noise of the suction of air through the leak.

The main bearings, crank pin or wrist pin bearings having too great a clearance and so allowing the lubricating oil to escape too easily, is another cause which can be proved by running the pump with the engine stopped and noting the quantity of oil running out from the main and crankpin bearings. Make sure that the cock on the lubricating line to the pressure gauge is open so that the gauge indicates the exact pressure.

Lubricating Oil Temperature Rises. If the outside of the tubes in the oil cooler are covered with rust or scale, it will yield less cooling effect, and produce a rise in the temperature of the lubricating oil, but if the inside of the tubes in the cooler are dirty, this will cause a rise in the oil pressure.

Another cause is tight bearings; newly fitted bearings must be watched closely and run for a short time only; then the engine should be stopped and all bearings examined for heating. Should the bearings be dirty or have grit in them, this will also cause heating of the lubricating oil, or the engine

may be overloaded, and the bearings begin to warm due to the excessive pressure on them. The bedplate, housing and cylinders should be felt occasionally for signs of overheating.

Lubricating Oil Temperature Drops. This means that the bearings are getting a good deal of oil for the load on the engine. No damage will result to the engine because of this. As the lubricating pump is often driven by the same motor that drives the cooling water pump it is run at the same speed as the other, the quantity of oil and water pumped being often controlled by regulating the suction or discharge valves. Too much lubricating oil does not cause nearly so much trouble as too much cooling water.

Circulating Water Pressure Rises. In this case some discharge valve may have jarred partly shut, or closed by mistake. The slowing down of the pump from this will cause a rise in the current used by the motor and can be noted by looking at the ammeter, if one is installed. Alternatively, there may be deposits of salt and scale in passageways for the cooling water of the cylinder and exhaust manifold. In this case the lines should be blown out and the cylinder jackets cleaned. Sometimes a zinc plug may work loose and get adrift and obstruct the passage of water, in which case it is often hard to find.

Circulating Water Pressure Drops. Sometimes a suction valve may jar partly shut or be closed accidently, or the suction strainer may be clogged up. In this latter case it should be blown out if there is an air connection fitted to the strainer; if not, the strainer must be removed and cleaned.

The wear on the pump rotor or impeller, may produce too much clearance and the pump will not keep up its full pressure. The pump should be taken apart and examined and the clearances checked up, and if necessary a new rotor or impeller fitted. At the same time the packings should be gone over and the pump otherwise put in good order.

Circulating Water Temperature Rises. If certain cylinders or heads become hot, it is certain that insufficient water is being circulated through that cylinder line. Disconnect water inlet to that cylinder and start the pump to see if the delivery is clear. Then stop pump and connect water inlet and dis-

connect water outlet from the cylinder head. If there is not a free flow of water from the cylinder head when the pump is again started it will be necessary to remove the handhole plates to the cylinder jackets and clean them out as they are probably choked with scale.

All cylinders should receive the same cooling effect with the valves to the cylinder jackets wide open. Changes in the quantity of water supplied to each cylinder can be adjusted by opening or closing these valves. If it becomes necessary to keep all the cylinders nearly shut off with the exception of one or two in order to keep them reasonably cool, it is time to find out why those two cylinders are so hard to keep cool.

Certain cylinders may get air or steam bound. This is usually due to air getting into the circulating system, either by the pump sucking in air or to a leak in the air coolers. Both of these causes will allow air pockets to form in certain high points in the cylinders and heads, where there will be no water. These points will get very hot, and what little water that does reach these points will be turned into steam. This trouble can be remedied by fitting small pet-cocks in the high parts of the water jackets and by opening them occasionally the trapped air will escape.

Water Temperature Drops. This is caused by the circulating water pump running too fast for the particular speed and load of the engine. There is a certain temperature that an engine will run best, and the pump should be so regulated that the cylinders are kept at this temperature. A cold engine gives poor fuel economy due to failure to burn the fuel properly and to excessive heat losses to the cooling water.

Air Compressor Troubles. With the compressor working properly and the spray air system and spray valve packings tight and in good condition, it should always be possible to hold the same pressures on the gauges of the various stages with the spray air control lever in the same position at a given engine speed. If, as time goes on, these gauges gradually change when this lever is in the same position, it is a sign that somewhere in the system there is a leak starting, or that something is beginning to wear out. If, on the other hand the gauges should suddenly change, it is an indication that

something has either blown out or a valve has given way. In the following paragraphs are given a few of the most common troubles of the compressor and spray air system of the Diesel engine.

Air Pressure Drops on All Gauges. Care should be taken to prevent leaky joints in the spray air system or in shut off valves. If the engine is stopped an air pressure from a starting tank can be put on the spray air system and such leaks found by the escape of air.

A leaky first stage suction valve will heat up the intake pipe near it considerably and should be discovered before trouble arises, if the pipe is "felt" regularly by the attendant. Take out these valves, examine, clean and grind them in. See that the gaskets under the valve cages are in good condition when replacing the valves.

Loss of pressure may be caused by broken or stuck first stage piston rings. This could not be the case with air compressor pistons such as those shown in Fig. 55-D, as there are no real first stage rings; these are the second stage rings between the first and second stage cylinders. With these compressors, broken or stuck piston rings below the second stage piston will cause a drop of pressure on all the gauges.

Another cause is too great a clearance on compressor piston at top stroke. The clearance of any stage is found by inserting a piece of lead wire into the cylinder through one of the valve cages. When the engine is jacked over, the lead wire will be compressed between the piston head and the cylinder and the thickness of the lead wire, when withdrawn and measured with a micrometer, will be the clearance between the piston and the cylinder when the piston is on top stroke. If there is too much clearance, according to the design of the compressor, it is probably caused by worn bearings. This excessive clearance may be corrected by shimming up the connecting rod.

Should the compressor intake become partly stopped up, this will have the same effect as partly closing the spray air control lever. The gauge shut-off valves should be adjusted while the engine is running to prevent the gauge hands from

vibrating too much. However, if they are closed too far they will not show a true reading of the pressure.

Air Pressure Rises on All Stages and Relief Valves Pop. If there is an extra high pressure showing on all gauges, or relief valves on the air lines begin to lift, which they should not do normally with the spray control lever in the proper position for the engine speed and load, it can be caused only by something stopped up, or shut off, in the spray air system.

Air Pressure Drops in a Certain Stage and Rises in the Stage Below It. The suction or discharge valves of that stage are leaking or are hung up. If they are actually hung up, the relief valve on the stage below will usually also pop, in addition to showing a rise in pressure. Feel the suction valve cage and pipe; if they are warmer than usual, it is probably due to a leaky suction valve which allows the pressure to back up into the next lower stage. The discharge valve and pipe are always very hot all the time, but sometimes if the pipe near the point where it enters the cooler is not so hot as usual, it will mean that the discharge valve is leaking. If there is doubt which valve is leaking, take out both the discharge and suction valves of the lowest stage which shows the drop in pressure, and clean and grind them.

If the valves are found in good condition, the drop in pressure may be caused by the piston rings of that stage being either stuck or broken, allowing the pressure to leak back into the stage below it.

With some compressors there are a great many first stage suction and discharge valves, and it is sometimes very difficult to determine which one is leaking or broken without taking out all the valves. In this case it is best to remove them all and give them a thorough cleaning and grinding in.

Troubles From Lubricating Oil. Lubrication of the air compressor is very important. Too little lubrication will cause undue wear on the cylinders and pistons. Too much will cause gumming up of the compressor and spray valves, and possibilities of an explosion if high temperatures are accidently reached. Soapy fresh water has been used as a lubricant with more or less success; a small quantity being fed in through the first stage suction valves at half hourly intervals. Under

normal operating conditions the discharge temperature for two-stage air compressors is approximately 475 degrees F. Oil that is used in the air compressors which is vaporized at or below this temperature is carried over to the cooler, is there condensed, and may be drained off at frequent intervals if means have been provided for that purpose. At the same time there is a considerable portion of the oil which cannot be vaporized at the normal running temperatures. This part of the oil is deposited in the valves and in the discharge pipe. If the second stage discharge valve sticks open either due to the above deposit or defective springs, or mis-alignment, the air is recompressed without cooling and the temperature ascends to a point where that portion of the oil which has been deposited in the discharge pipes begins to evaporate. When a sufficient amount has been evaporated to form an explosive mixture with the air that is being recompressed, ignition will take place, resulting in an explosion.

Clearances between the piston heads and the cylinders must be kept very close to that arranged for in the design. Undue clearance will cause re-expansion of the undischarged air in the clearance space during the suction stroke and decrease the volumetric efficiency of the compressor. The usual clearances on three stage "steeple" type compressors is about .045 inch for the first stage, .035 for the second, and .030 for the third stage. First and second stage clearances are usually adjusted by placing or removing shims from under the foot of the connecting rod. Third stage clearance is adjusted by means of gaskets or shims under the valve cages.

CHAPTER XI

Representative Types of Engines

Worthington Two Cycle Solid Injection Engine—Worthington Double Acting Two Cycle Engine—McIntosh and Seymour Four Cycle Engine—Southwark-Harris Two Cycle Engine—Busch-Sulzer Bros. Two Cycle Engines—Busch-Sulzer Four Cycle Engines—Standard Fuel Oil Co. Two Cycle Engines—Allis-Chalmers Four Cycle Engines—Lombard Four Cycle Engine—Nordberg Two Cycle Engines—De La Vergne Four Cycle Engines—Craig Four Cycle Engines—Nelseco Four Cycle Engines—Fairbanks-Morse Type F. M. Solid Injection Engines—Bethlehem Large Unit Two Cycle Engine—Bethlehem Small Unit Solid Injection Engine—Nelseco Solid Injection Engines—Foos Four Cycle Engines

Worthington Two-Cycle Solid Injection Diesel Oil Engines. The Worthington, two-cycle, solid injection Diesel engine was the first American Diesel engine operating without compressed air for spraying the fuel oil, and without mechanically operated spray valves requiring accurate adjustment. With all its simplicity it still retains the essentially good operating qualities of the ordinary Diesel engine. Fuel oil is injected by a pump, the compression is high and there are no ignition or vaporizing devices. The combustion is non-explosive and free from detonating shocks. The fuel consumption is low, and the same grades of fuel oil are used as in the air injection Diesel engine.

In its machine design, the best practice is followed; the engine is of fine appearance, and strong in every part. All wearing parts are liberally proportioned to insure long life and the consumption of lubricating oil is remarkably low. Much of the mechanism common to older Diesel engines has been completely eliminated, and there are no delicate adjustments anywhere.

The most important feature in the working of this engine is the combustion chamber. This is designed so that the rate

of combustion in the cylinder is not the same as the rate of fuel oil injection by the pump. The pump injection is fast, but the combustion is slow and graduated, quite similar in effect to that in ordinary Diesel engines and but slightly affected by the pump timing.

This valuable feature of independence of rate of combustion in the cylinder with reference to rate of injection, is accomplished by new means consisting essentially of a divided, or two-part combustion chamber. Fuel is injected into the smaller part or injection chamber, which holds sufficient air to burn only a small part of the fuel during injection. This injection chamber is connected to the cylinder holding the main air and constituting the second part of the combustion chamber, through an ejection orifice. Cylinder combustion takes place only when the fuel and unused air flow out of the injection chamber, through the ejection orifice, and into the cylinder. The rate of combustion in the cylinder is the rate of ejection from the injection chamber, and this is controlled by the relative pressures on the opposite sides of the ejection orifice. Control of ejection by orifice pressures is fixed once and for all by the design of injection chamber, spray nozzle and orifice, and automatically maintained independent of fuel injection, without any moving parts, except the piston itself.

Combustion is complete and a large part of the air charge is utilized in the combustion, by concentrating all the air in the cylinder directly in front of the ejection orifice, and by the mixing action produced when the jet of fuel and air is projected through the ejection orifice into this concentrated air. Concentration of cylinder air is accomplished by using a flat top piston, and a cylinder head flat at the edges, so both faces being machined may come close together, thus pushing all air into a dome shaped space in the middle of the cylinder head with the ejection orifice in the top.

The symmetrical forms of cylinder head, injection chamber and piston top forming the boundaries of the combustion chamber, reduce heat stresses and tendency to crack, while contributing directly to efficient, slow and shockless combustion of heavy fuel oil.

In Diesel engines of earlier design major breakages of

such parts as crank shafts and cylinder heads were not infrequent, as users well know, nor have such troubles been confined to any particular line of Diesel engines. Worthington has taken extraordinary pains to develop a design that would reduce these two risks, and the design adopted does practically, if not actually, eliminate them.

Crankshaft breakages have been due, in the main, to two things, preventives for both of which are incorporated in this new Worthington product. First of these is misalignment of the shaft so that it rotates out of line, its axis being a curved instead of a straight line, and certain undetermined stresses of a reversing nature are set up, due to bends, so that in a given number of reversals, any shaft would break. Misalignment is prevented in this new design in two ways. In the first place, concentric bearing shells are inserted in the lower frame member in cylindrical seats, machined in such a way as to absolutely insure the original alignment. No adjustment is provided that might be wrongly made by an unskilled operator, except the usual one of taking up wear by screwing down of bearing caps after removing shims. This operation would throw the shaft out of line if the wear were unequal, but equality of wear is provided for also in this engine by means that practically prevent all wear, the forced circulation of excess quantities of lubricating oil through all bearings. A closed crank case prevents contamination by dust or water. This oil is delivered by a pump forming part of the engine and is circulated in far greater quantities than the lubrication requires, so that there is always an excess to maintain a pump created film supporting a shaft practically independent of the viscosity of the oil. To maintain a steady state where there is any tendency for this oil to get warm, a water cooled oil cooler is inserted in the circuit.

The second cause of shaft breakages is the momentary and periodic overloading or stressing of the shaft by detonating shock-like pressures developed in the cylinder during combustion. If the oil, when injected, is not suitably controlled with reference to its rate of combustion, conditions may rise in which the combustion may be too rapid, approximating a true detonation and developing a shock on the running gear

that is so severe as to be accompanied by a pounding or knocking noise, easily audible. Under certain conditions of operation, any oil engine may occasionally produce such a shock; for example, during the starting period, but experience has indicated that such cases do no harm. The harm comes, in the form of shaft breakage, when the shocks are regularly recurring, though not necessarily continuously.

Prevention of such shocks is one of the main ideas in this new design and one of the ideals which experience proves has been successfully attained. The means whereby these shocks are prevented are new and are so sure as to receive the assent and acceptance of everybody that has so far studied the situation. It would be clear, from a moment's consideration, that no shock at all could be produced during injection if the oil entering could not receive enough air to burn any substantial part of the oil. This feature is incorporated in the design of the divided combustion chamber which prevents the recurring shocks that contribute to shaft breakages.

The second great class of breakages, of cylinder head failures, has also been met by designing preventives in this new Worthington engine. There are two reasons for these cylinder head breakages in the past; one of them is the shock pressures noted above in connection with crankshaft breakages and the other is expansion stresses set up by reason of the fact that the inner side of the cylinder head becomes hot and the outer wall between the jacket water and the atmosphere stays cold. This difficulty of expansion has been met in the new design in two ways. In the first place, the cylinder head wall in contact with the burning gases is made symmetrical and of an expansion shape having a flat ring with an inner dome of approximately half the cylinder diameter. This will, by its shape, take up its own expansion, but even this is not relied upon alone in this new cylinder head design. Another contributory element is added, and that is a two-part construction of the cylinder head. The top wall of the cylinder head is not cast in one piece with the bottom wall, the assembly of the complete head involving a gasket joint which is necessarily an expansion joint. Furthermore, the casting of the upper and lower walls of the head separately permits

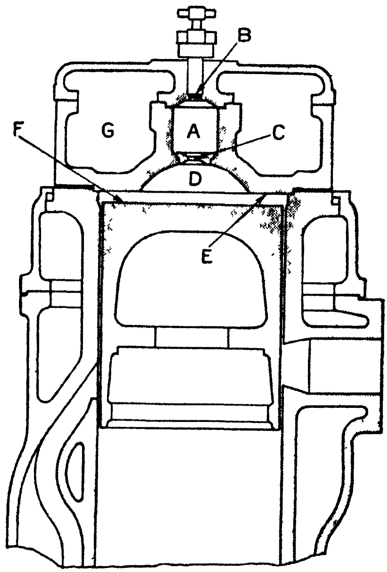


Fig. 146.—Showing Injection Chamber of Worthington Solid Injection 2-Cycle Engine

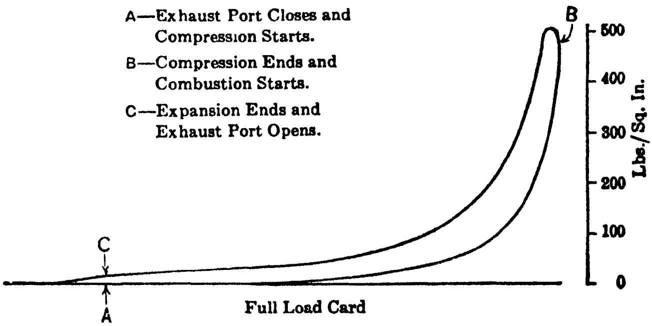


Fig. 147.—Indicator Card from the Worthington Engine

an inspection of all metal thicknesses and of the water space, so that there is absolute insurance that no intended water space has been filled up in the foundry by core sand or metal. This construction makes as nearly an unbreakable head as it is possible to devise.

In the center of the cylinder head, Fig. 146, a small cylin-

drical space is formed. This space is the injection chamber, its top being closed by the upper half of the cylinder head carrying the sprayer "B." Into the bottom of the injection chamber, a removable circular plate "C" is screwed, to provide constantly open communication with the cylinder, and to permit gases to flow in either direction as a smooth jet whenever the pressure on one side is higher than on the other. This ejection orifice is machined to size, so as to control the pressure differential to the best values for up flows of air during compression, and down flow of fuel and air during combustion, and finally, of air alone during expansion.

In the dome space "D" between the flat top of the piston and the under side of the cylinder head all of the air in the cylinder is concentrated directly under the ejection orifice. The bottom of the lower half of the cylinder head is machined flat at the edges "E" so the flat machined top of the piston "F" can be brought very close without danger of touching.

At the end of the compression stroke the whole cylinder charge of air, divided between the space above the piston under the head and the injection chamber, has been raised in pressure to about 475 lbs. per square inch in the cylinder, and something less, say 450 lbs., in the injection chamber. The ejection orifice is a restriction for the air flow from cylinder to injection chamber during the up stroke. All the air has been heated by compression to a temperature much higher than the ignition temperature of fuel oil, so oil injected will at once begin to burn.

The water jacket "G" surrounding the cylinder head and injection chamber keeps all walls cool, but the air being rapidly compressed, is not chilled enough to prevent prompt ignition because of its low heat conductivity.

At the end of compression, the injection chamber and cylinder combustion chamber are, therefore, full of compressed air hot enough to ignite fuel. Most of the air is in the cylinder, only a lesser part being in the injection chamber.

Just at this time, or more exactly at about 16 degrees of crank angle, before top dead center, the fuel pump starts injection of the fuel downward from the sprayer "B" and across the injection chamber, directly toward the ejection orifice

ring. The fuel injection continues until the governor, by causing the by-pass valve to open, cuts off fuel flow to the cylinder.

As soon as the spray enters the injection chamber, combustion starts, but due to the small surface exposed, only a little oil can burn, just as much as actually touches the air and this is controlled by the design. The spray cloud, with vapors and products of partial combustion, finally locates in the bottom of the injection chamber. Only a small part of the oil has burned, so the pressure rise is only a fraction of what it might otherwise be. All of the air in the injection chamber at the top and sides is not used in this first stage of combustion, which takes place during injection. This air plays its part later.

At the end of the injection period, which is also the period of first stage combustion, and just about dead center, the pressure in the injection chamber has been raised from something less than the cylinder pressure to something greater; just how much greater depends entirely on the design of the spray plate and injection chamber.

This tends to reverse the flow through the ejection orifice from air flow upward to gasified fuel flow downward. This down flow is accelerated by the outward movement of the piston, slowly at first and then at a rapidly increasing rate, making a downward jet of fuel burning in the cylinder faster and faster till it is all consumed.

To promote the complete combustion of this self feeding fuel jet, several factors are made to co-operate, each contributing its part. The first is the concentration of cylinder air into the central dome chamber, so the fuel does not have to travel far to reach the air. The second is the agitation of the cylinder air and gases by the downward jet spreading out sidewise, helped by the natural tendency of the center to flow toward the edges as the piston leaves the cylinder head. The third is the expansion of the air left in the injection chamber and not used in the injection and pre-combustion period. This air, following the fuel jet into the cylinder, helps to burn it and keeps up the cylinder agitation or turbulence, even after all fuel has passed from the ejection orifice into the cylinder.

This second stage of combustion, which is the main com-

bustion, takes place in the cylinder under ideal conditions for complete combustion. Its rate is controlled mainly by the piston which determines the pressure differential on the ejection orifice and therefore the rate of fuel ejection. Combustion is therefore not only complete but automatically regulated. It is non-explosive in type and the rate is obviously independent of the time of pump injection, which latter therefore need not be accurately adjusted.

Having traced the sequence of events from the start of fuel injection and first stage combustion to the end of the second stage, or final combustion, the other events in the cylinder can be followed and their relations clearly seen from the indicator card, Fig. 147. Here the rise of pressure during compression on the up stroke from one atmosphere at the start "A," after the exhaust port has been closed by the piston, is indicated by the line "A-B." This ends at "B," where the compression pressure is about thirty-three atmospheres, enough to make the fuel sure to ignite. At "B," injection of fuel starts and results in a slight rise of pressure, due to the first stage combustion in the injection chamber. Then as the piston starts to move down, the pressure instead of falling as it would if there were no further combustion, is at first prevented from falling at all and then falls slowly as ejection becomes active and the second stage, or main combustion in the cylinder proceeds. Combustion ends at a point difficult to exactly locate, and expansion starts, continuing till the exhaust port opens at "C."

It is clear from this indicator card that there is at no time any sudden change of pressure, so that there are no shocks such as are common when combustion is explosive, as is the case in some engines. The cylinder pressures always exert a downward force on the piston on both up and down strokes. The indicator card of pressure is almost identical with that of the unaflow steam engine and operation is equally smooth and regular.

The whole series of events in the cylinder producing the changes of pressure with up and down stroke of the piston shown in the indicator card, Fig. 147 are made more clear by the diagrams of the piston positions in the cylinder, Fig. 148.

In the left-hand diagram "A," the piston has started up and has just closed the exhaust port on the right, starting the compression. At the top of the stroke, diagram "B" shows the position for fuel delivery by the pump into the injection chamber, producing the first stage of combustion. This is followed by the second stage of combustion in the cylinder, ending when the piston is about in position "C," at which time expansion starts. Expansion ends in position "A," piston moving downward, and exhaust starts. Further downward movement of the piston uncovers the air intake or scavenging port on the left, position "D." The air, at between three and four pounds per square inch pressure, from the scavenging

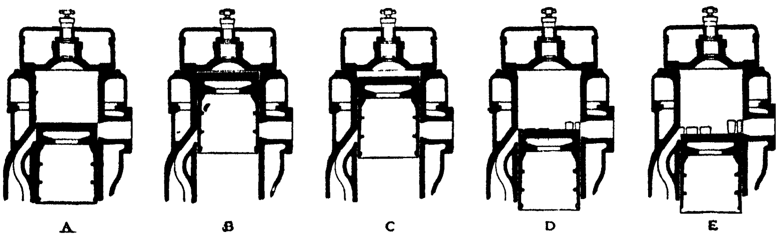


Fig. 148.—Cylinder Ports and Piston Positions, Worthington Engine

pump enters in a series of upward streams and displaces products of combustion out through the exhaust ports. This air admission is guided on one side by the inclined wall of the port and on the other by the vertical face of the piston, so that it moves almost vertically upward to the cylinder head around the lefthand side of the cylinder, while burnt gases are being displaced downward and out of the exhaust ports around the right-hand half of the cylinder. This scavenging action by means of which the cylinder is charged with fresh air, starting at position "D," continues through position "E" and ends at position "A."

To supply the slightly compressed air required for charging the cylinders of all sorts of two-cycle engines when the piston uncovers the scavenging or air intake ports, there are in use five standard arrangements, besides some special ones. Proper selection of the one best suited to the purpose for

which the engine is designed, is an important factor in the success of the engine.

The Worthington scavenging arrangement follows the general structural lines of all large Diesel engines, in having the closed crank case with its circulating forced feed lubrication of main running parts and independent cylinder lubricating system, with crosshead, stuffing box and piston rod. This makes it a simple matter to utilize the crank end of the cylinder as a scavenging pump, and to provide the necessary air receiver space between the crosshead guides and the frame

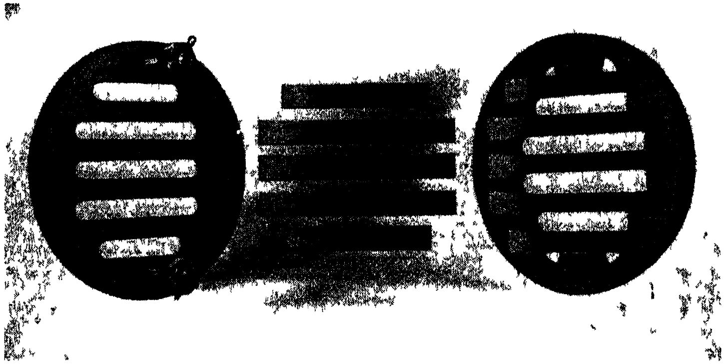


Fig. 149.—Worthington "Feather" Valves

and around the base of the cylinder, keeping all air flow out of the crank case, which is tightly sealed.

On the upstroke of the piston, air is drawn through passages for each cylinder cast on the frame, to the air receivers behind the cross-head guides and into the bottom of the cylinder. These valves are known as "Feather" valves and are shown in Fig. 149. These feather valves allow the piston to draw in its air charge with the least resistance, and being straight spring steel strips, which flex against cast iron faces in both open and closed positions, their reliability is unequalled. On the down stroke of the piston the air is compressed by the piston displacement into the receiver and connecting spaces, including the cylinder intake or scavenging port. These spaces are made large enough in proportion to

the piston displacement to give the desired pressure of 3 to 4 pounds per square inch above atmosphere for quick charging of the cylinder when the ports are uncovered, but with the least power absorbed in supplying the air for scavenging.

Air entering the cylinder is directed upward and away from the exhaust ports by a new design of inclined air scavenging port co-operating with a flat top piston. This more effectively displaces burnt gases without loss of air than older forms of straight ports requiring ribs in the piston to deflect the air upward.

Cracking of cylinder heads and pistons, with burning of pistons also in some cases, are well known defects of older oil engines that have been so common as to lead many to believe that cracks are to be expected and cannot be prevented. Study of the causes of these cracks indicates that while there are undoubtedly limits of operation beyond which cracks or burns cannot be prevented in the very large cylinders, or at excessive speeds in smaller ones, prevention is mainly a matter of good design and material. Cracks of this sort are always due to heat and are the effects of excessive inequalities in the temperature of different parts of the castings in operation. There are two general types of remedies independent of selection of material. One is to limit the temperature difference in the parts of the metal to a value low enough to avoid so much distortion as will produce a crack. The other, and more important, is to select such shapes as will expand uniformly and freely on heating without concentrating the expansion stresses at any point. Worthington has incorporated both of these remedies in the cylinder heads and pistons of this engine to a greater degree than has been possible heretofore.

The essential thing is a symmetrical form. By using perfectly symmetrical shapes, the metal expands freely everywhere and this free metal expansion is assisted by making the head in two parts, so the cold top wall does not restrain the lower hot one. This is an exclusive Worthington feature, shown in Fig. 150 and in the sectional views of this engine. It also permits of the most perfect cylinder head jacket, because when the two parts are separated, the whole jacket space is exposed for inspection and removal of mud or bad

water scale, which would in time produce the over-heating that results in cracks even in the most perfect design.

Worthington indirectly cooled pistons used in the smaller engines are designed in a special way that has proved in service to be of great value. Not only is the principle of

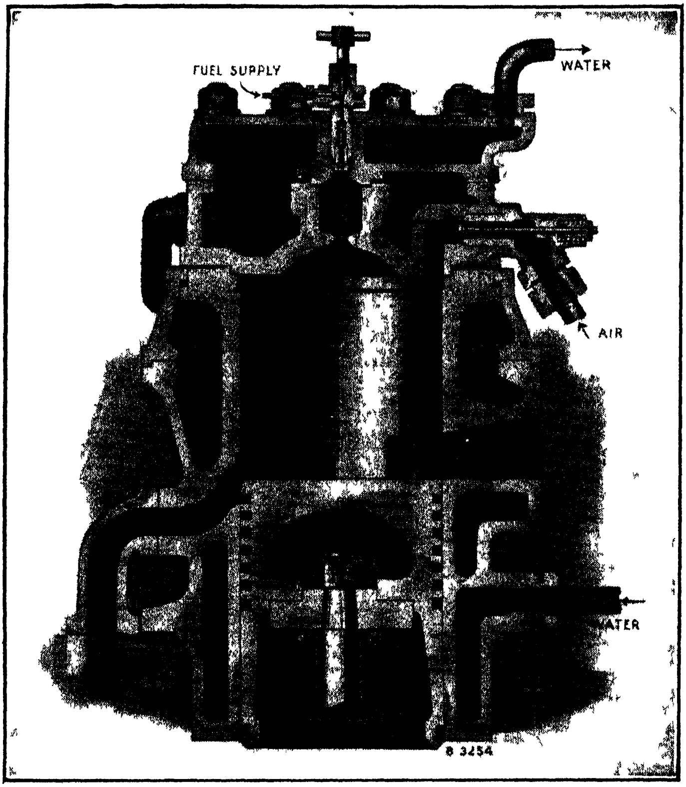


Fig. 150.—Sectional View of 30 H.P Worthington Cylinder and Cylinder Heads and Immediate Attachments

perfect symmetry carried out, as already explained, but an additional principle is incorporated in the interest of avoiding cracks and burns peculiar to indirectly cooled pistons. This is the thick piston idea, which prevents any part of the mass of piston metal getting materially hotter than any other part, by providing sufficient metal to conduct the heat. Heat

is received by the piston all over its head and somewhat more intensively in the center than at the edges, due to the fact that fuel enters the cylinder downward at the center. The heat so received, flows radially outward through the piston metal and then downward along the piston barrel where it crosses the lubricating oil film to the cylinder wall and finally reaches the cylinder jacket water.

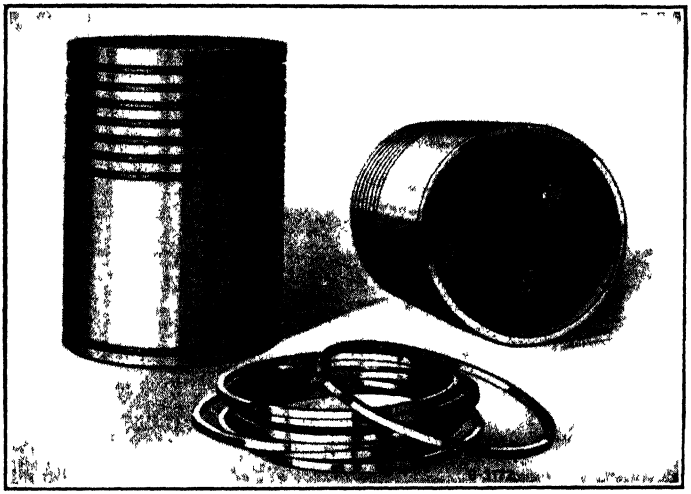


Fig. 151.—Worthington Piston and Rings

This heat flow indicates the necessity for graduating the metal thickness in proportion to the amount of heat passing through any section exactly as would be necessary for water flow passages or conductors of electricity. The maximum thickness is required at the edges of the top and the upper part of the barrel, and easy curves without ribs are used in the interests of symmetry and good foundry practice.

The external appearance of the piston is shown in Fig. 151 with its seven narrow snap rings for tightness, without excessive cylinder wall friction. On the same view is shown the internal ledge provided with studs to receive the piston rod flange.

In the larger sizes of cylinders the heat is generated at a

rate too great for any indirectly cooled piston, and in these cases the best standard Diesel practice is followed, by providing each piston with a circulating oil jacket. This is fed through the piston rod by a large excess of lubricating oil which is collected and cooled as will be clear from Fig. 152.

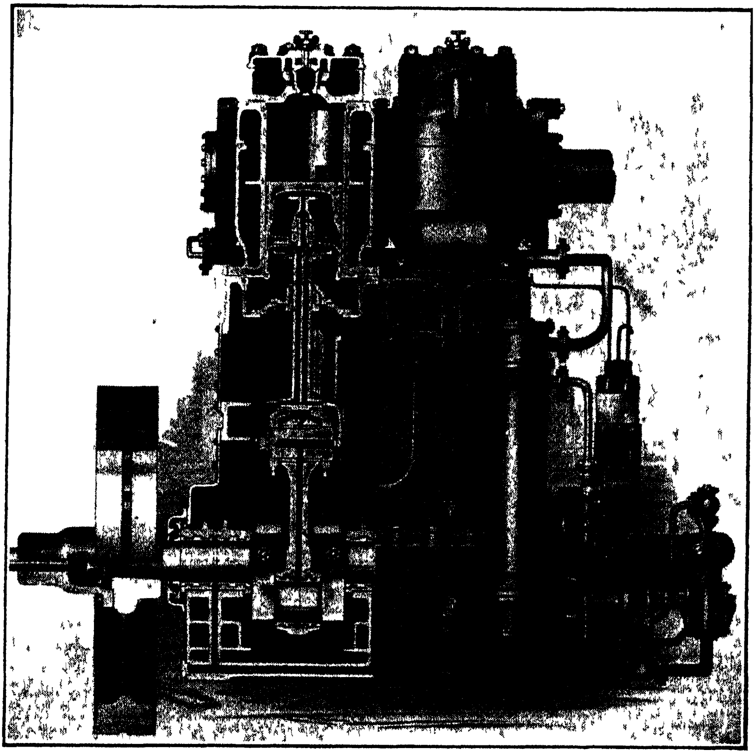


Fig. 152.—Longitudinal Section of Worthington Engine

Lubricating oil is used for cooling the piston for three reasons. First, so that the same pump will do for delivering both lubricating and cooling oil to the shaft crank and rod passages, which makes the supply system no more complicated than for any good forced feed circulating system of lubrication. Second, to avoid the complication of closed pipes for discharging heated oil from the pistons, and permitting the use of a

simple open orifice attached to the crosshead, discharging the oil into a chamber directly under the orifice. Third, to permit the use of one cooler for both the lubricating and the piston cooling oil.

The cylinder is a specially designed casting, fully water jacketed, carrying air scavenging and exhaust ports, with provisions for a frame connection insuring alignment with crosshead guides and for head connection insuring both gas tight and non-leaking inter-jacket joints. The jacket casing for the upper end of the cylinder is sectionalized as shown in Fig. 150, to provide an expansion joint between inner hot walls and outer cold walls.

The lower end is flanged and faced for frame studs and is provided with a centering projection fitting a bore on top of the frame which is machined at the same tool setting used for boring the crosshead guides. This insures that the cylinder will always be in line, no matter how often it may be removed and replaced. Several large cleanout plugs are located at the bottom of each jacket for working out mud from bad water. The top section of the cylinder jacket being removable, not only acts as an expansion joint, but is also a mud and scale clean out opening, giving as free access to the cylinder jacket as does the two part construction of cylinder head to the cylinder head water spaces. The jacket at the cylinder top is closed and circulating water passages to the cylinder head at both front and back are provided through suitable connections solidly bolted to both cylinder and head, and designed to prevent any possibility of water getting into the cylinder.

A scavenging air port is cored into the cylinder through the water jacket on an upward incline. A passage is provided to connect the cylinder port with the scavenging air chamber.

The main running parts of the engine are shown in Fig. 153, assembled and with separate views of some parts. The assembly also shows the special stuffing box for the piston rod, and which is carried in a removable plate at the crank end of the cylinder, made fast to the top of the frame and constituting the top closure of the closed crank case.

The piston rod is a steel forging, flanged at the lower end

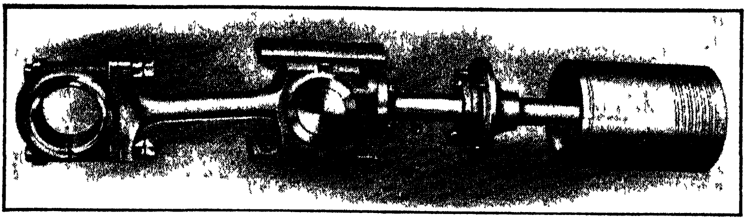
for bolting to the crosshead. At the upper end it is fitted on a taper with large shoulders to a heavy flange bolted to the piston. This rod is practically always in compression, so the fastenings are but lightly stressed. The removal of the nuts securing the piston rod to the crosshead and of those securing the stuffing box to the cover at the upper end of the frame permits the removal of the piston, piston rod and stuffing box through the top of the cylinder.

The crosshead, Fig. 153, is of the double bored guide type, the bearing surfaces being large so that the pressure per square inch is at all times low, and the guides are so designed and machined as to insure accurate alignment of the crosshead with the cylinder and the crank pin. Slippers are provided with shim adjustment for wear and so constructed as to prevent loosening and jamming in service. The wrist pin is force feed lubricated, as also are the guides.

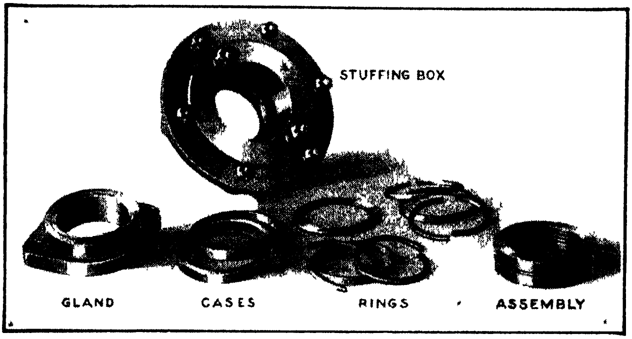
The connecting rod is a high grade steel forging with marine type babbitted box on the crank pin end, and solid bronze bushing on the crosshead end to avoid any chance of loosening. With the forced feed lubrication used, wear is practically eliminated. A hole carries the pumped oil from crank pin to wrist pin.

The crankshaft forging for engines having four cylinders or less is made in one piece. For engines with more than four cylinders the shaft is usually in two pieces, rigidly bolted together. One end of this shaft carries the driving members of the control box and the other end is tapered to take the flywheel hub to insure accuracy of fit and alignment. On the flywheel taper a key is fitted to take the torsional stress, and the outer face of the flywheel hub is machined to take the match coupling of the shaft extension. Oil throw-off rings are provided near the outer end, at the frame housing, to prevent creep of lubricating oil which is pump fed to each main bearing. Holes drilled in the shaft, cranks and crank pins carry oil to the connecting rods.

The base, for all engines up to four cylinders, or less, is a single casting, while those having over four cylinders are made up of two similar castings solidly bolted together end to end into one rigid piece.



Reciprocating Parts of Worthington Engine



Stuffing Box and Parts

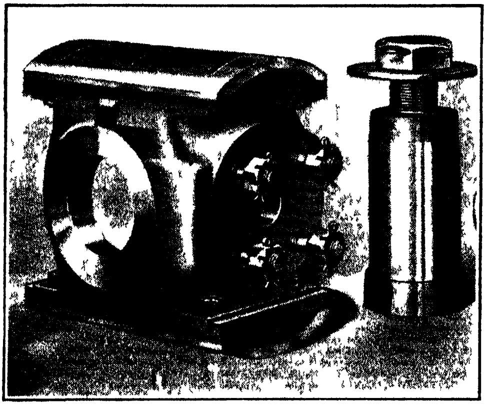


Fig. 153.—Worthington Cross-Head and Wrist Pin

This base casting is machined on top to receive the frame, on one end to receive the control box, and on the other to take a lubricating oil collecting box receiving oil from the throw-off ring on the crankshaft. The base is also bored and faced

to receive the concentric main bearing shells which are babbitted and which can be removed and replaced by rotation after the shaft is slightly raised. This construction insures alignment of the crankshaft. The vertical construction and use of forced feed circulating lubrication of these bearings makes the wear so slight and always equal on all bearings, that the original alignment is retained in service.

The lower part of the base casting has formed in it a lubricating oil sump depression, to which reference will be made later in connection with the lubricating system.

In line with base construction, the frame is also a one piece casting for engines having four cylinders or less, while for engines with a greater number of cylinders it is usually made in two sections, each carrying half the number of cylinders required for the complete engine. This frame, when bolted to the base with sufficient fitted location bolts, in addition to the unusually large number of connecting bolts, makes a most rigid structure with alignment of all parts absolutely assured.

In front of each center main bearing an opening is provided front and back, and this is closed by a stiff cast iron cover bolted in place with gasket to prevent oil leaks. At each end there is an angle plate, one face of which bolts to the frame end and the other to the base to cover the end main bearings in operation, or to give complete access for inspection and adjustment.

Just inside of the top of the frame a web is cast and later machined to a fit, and bored to receive the lower or crank end of the cylinder. Central location of this is assured by machining these surfaces at the same setting used for the others.

This frame structure is remarkable in many respects. Being made of the minimum number of parts, and accurately machined, alignment of cylinder and piston is assured with reference to crosshead guides, and in turn their alignment with reference to crank pins and main bearings is also assured. Coupled with this most unusual assurance of alignment of all running parts, is a correspondingly unusual degree of stiffness to resist distortion that makes rubbing surfaces wear,

and complete accessibility of all working parts. This combination of assured alignment of parts, stiffness and accessibility with the closed crank case necessary for perfection of circulating forced feed lubrication, has seldom been equalled and never exceeded in any oil engine.

The scavenging air suction valves are located at one side at the top of frame and covered with silencing orifices and,

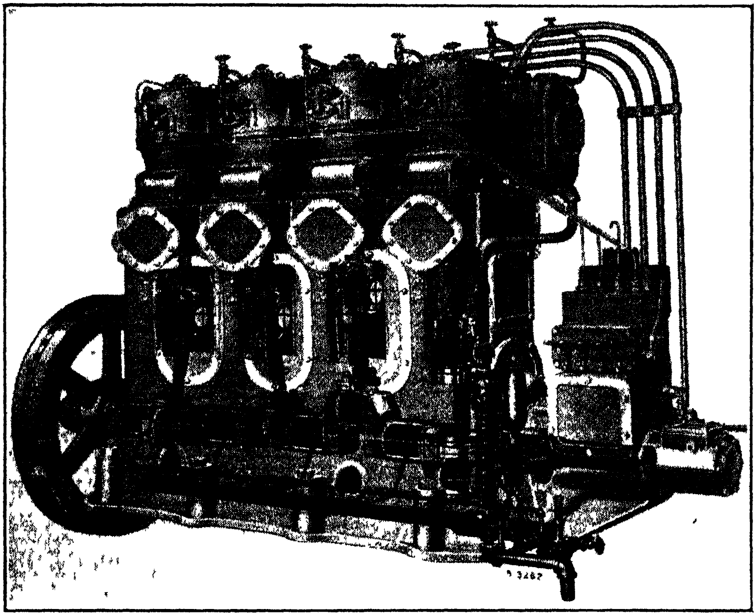


Fig. 154.—Circulating Forced Feed Lubricating System for Main Running Parts of Worthington Engine

at the other side, are located the air transfer ports, leading air from the receiver to the cylinder inlet ports.

The main lubricating system, Fig. 154, entirely independent of the cylinder system, is automatic in its action. In small cylinder engines with indirectly cooled pistons a single plunger pump driven by an eccentric takes oil from the engine base sump through a strainer and oil cooler and delivers it to a header on which a gauge indicates the pressure. This pressure is kept constant by the action of the pump which has a

capacity many times greater than all bearings combined really need. In large cylinder engines using oil cooled pistons a large capacity gear pump supplies both lubricating and piston oil.

From the main lubricating oil pressure header a branch is led to each main bearing which is drilled to feed the oil to a circumferential groove. From this groove around the middle of each main bearing oil flows three ways, toward each end of the bearing and to a hole drilled in the shaft. Oil escaping from ends of the inner bearings drops directly into the crank pit. Oil escaping at the flywheel end is thrown off by a throw-off ring on the shaft into the collection cover box bolted to base and frame, from which it drains back to the crank pit. Oil escaping from the other end feeds the control box, from which it overflows back to the sump, keeping a constant level in the control box.

Oil entering the shaft holes at the center groove of each main bearing, flows to the crank pin, where the triple flow repeats itself, and the mainstream continues up the connecting rod, lubricating wrist pin and both crosshead guides, all oil dropping back to the sump finally.

This arrangement is standard for engines on which the smaller cylinders are used. For engines carrying cylinders of the larger sizes, in which the pistons are oil cooled, the oil is carried from the crosshead up through holes drilled in the piston rod to a chamber in the piston. From the piston it returns through another hole drilled in the piston rod to the crosshead, from which it is discharged through a nozzle into a slotted pipe.

This slotted pipe carries the heated oil direct to the sump and prevents it from coming into contact with the running gear. The circulation of lubricating and cooling oil is therefore continuous from the sump through the pump, filter, cooler, running gear and piston, back to the sump.

Bearing heat is carried off by these never ceasing oil streams at each rubbing surface as fast as formed, so bearings are no hotter than the circulating stream which is cooled by a suitable water coil.

Losses of oil from the crank case are prevented, first, by

the gasket joint cover plates bolted fast; second, by keeping all air from the crank case which also keeps out dust and moisture, and third, by piston rod stuffing box in the top crank case cover plate. The interior of the crank case and control housing are well cleaned and painted with white enamel to prevent scale or rust from mixing with the oil.

This piston rod stuffing box is of special design intended more to prevent loss or contamination of lubricating oil than prevention of air leakage, since the pressure of the latter is only 3 or 4 pounds.

To prevent this, Worthington piston rod stuffing boxes are fitted with special packing to limit the oil carried up. To catch the small remaining amount scraped off on top, a collecting chamber is provided above the packing. This chamber is extended upward high enough to prevent scavenging air blowing past it from right to left carrying oil away, and it is in constant open communication with the crank case to drain back all oil collected. The pressure above while pulsating, always averages something higher than that in the crank case so the return of oil is positive and assisted by the flow of a small stream of scavenging air downward.

Cylinder lubrication begins in a multi-plunger sight feed lubricating oil pump with separate feeds for each cylinder. Oil is delivered by a plunger to each of two or four oil pipes, depending on cylinder size, ending at check valves on opposite ends of the same cylinder diameter and at such a level on the cylinder wall as is never uncovered by the piston, the length of which is made greater than the stroke. At the interior outlet of these two oil holes on the inner face of the cylinder a circumferencial groove is cut in the wall so that the oil feed is distributed evenly all around the piston. Cylinder oil is regulated so that there is normally no cylinder drip, all oil working out the exhaust port as it is used up. Should an excess be fed so that some drops down, it is caught by the lower cylinder head or frame cover and directed down to the scavenging air receiver to a point where a drain is connected to a collecting pocket. Appearance of cylinder oil at this pocket is indication of an excess being used. None can get into the crank case.

All engines are fitted with oil coolers of suitable construc-

tion which insure that all lubricating oil is properly cooled before entering the engine.

All small running parts that correspond to the valve gear of an ordinary engine are concentrated at one point in this

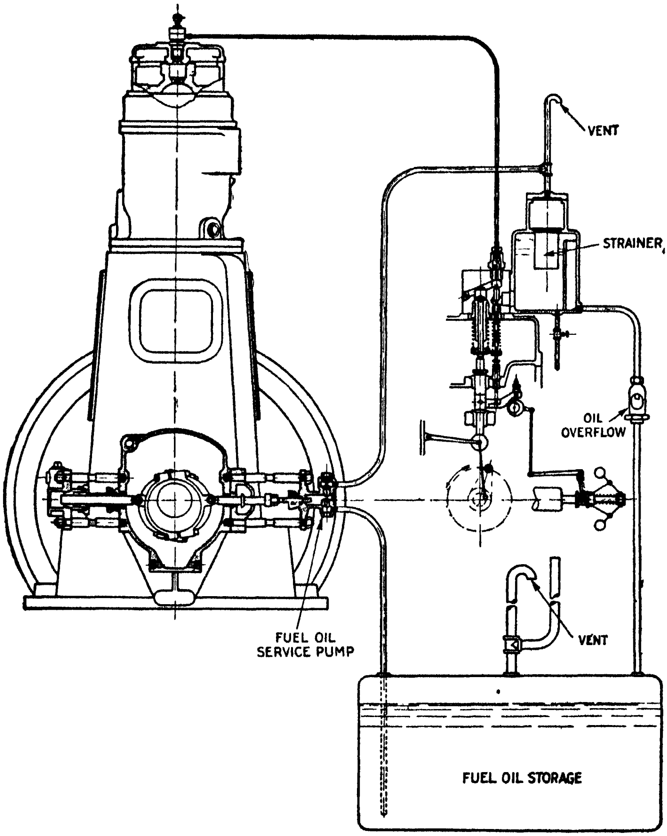


Fig. 155.—Diagram Showing System of Fuel Distribution, Worthington Engine

Worthington engine and enclosed in an automatically lubricated dust proof box bolted to one end of the base, and enclosing the end of the crankshaft.

The fuel system as a whole is shown in Fig. 155, from which it is clear that the two parts are independent, except

as to the injection suction box, which is common to both parts. This suction box is always kept full and contains enough oil to start the engine, even when the circulating suction pipe to the storage tank is empty. A few turns of the engine will serve to put the circulating system in operation before the

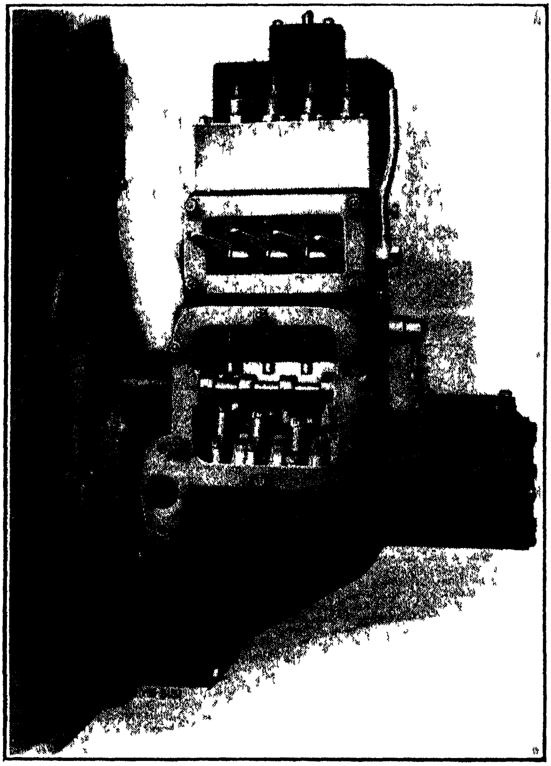


Fig. 156.—Control Box of Worthington Engine, Front View

injection pump suction box has emptied. This makes it unnecessary to use a hand pump to prime the injection from the storage tank. The fuel oil circulating pump is of simple horizontal plunger type, driven from an eccentric on the crankshaft in the control box and close to the engine base. This is the same eccentric that drives the main lubricating circulating pump on the opposite side.

Fuel oil enters the closed suction box at the center of the top under which point a cylindrical wire mesh strainer is located, attached to the cover plate. Excess oil not taken by the injection pump overflows from the side of the top of the suction box back to the storage tank. A vent pipe carries off any gases or vapors separated out of the oil in the suction box.

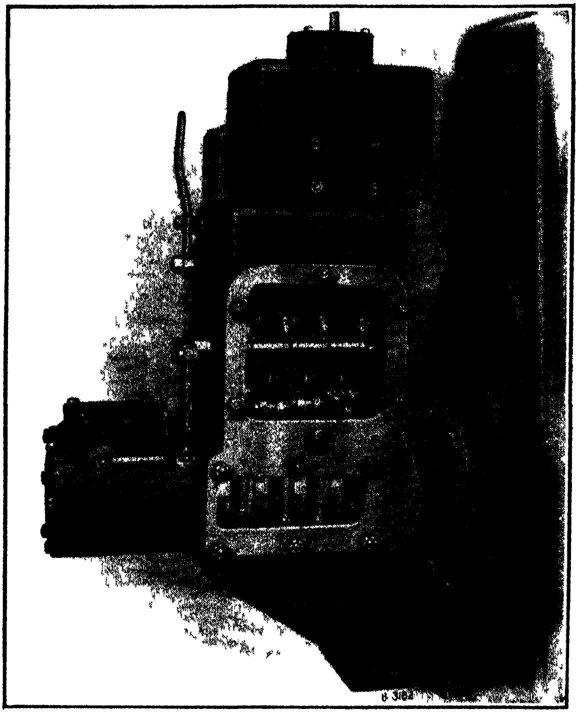


Fig. 157.—Control Box of Worthington Engine, Back View

This separation is assisted by the vibration and keeps gas bubbles out of the injection pump. Such gas bubbles would interfere with the accuracy of the injection pump metering, and also with the timing of its delivery to the spray nozzles.

Modifications have been standardized for use with fuel oil that is too viscous to be pumped and which therefore may require a little or considerable heating.

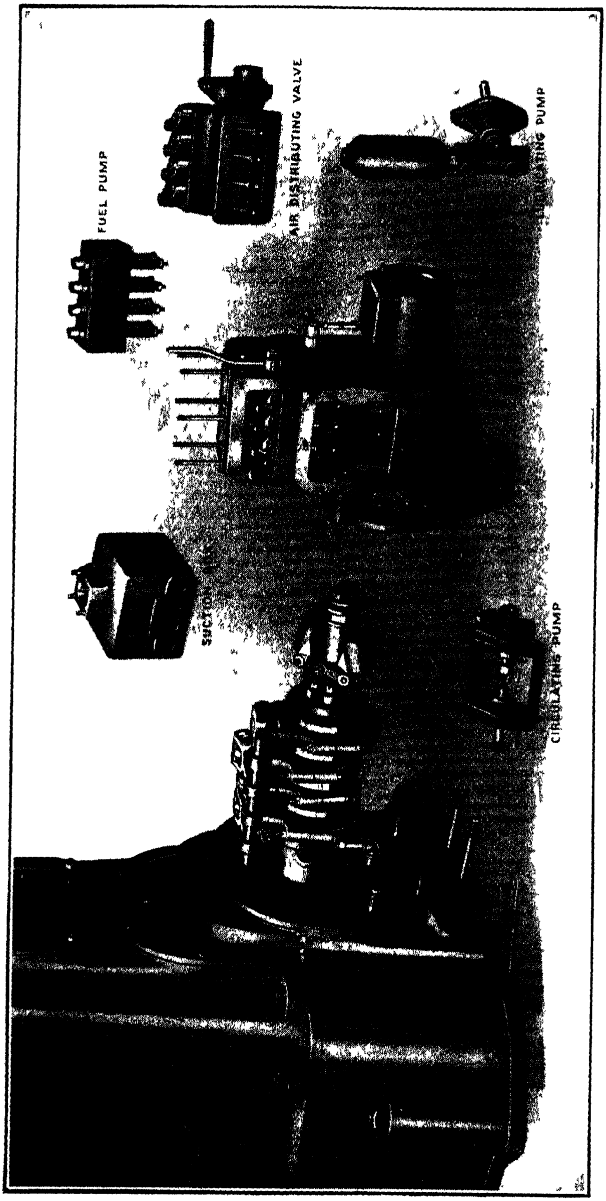


Fig. 158.—Control Box Parts, Air Starting, Fuel Service and Injection Pumps, Lubricating Oil Pump and Governor, Worthington Engine

Settling of solid matter in the suction box is promoted by its size and construction and a drain is provided for removing such sedimented dirty oil. By locating the pump suction port above the bottom of the suction box, the drawing in of dirty oil is prevented. At the same time, it is low enough to insure that the port is always submerged.

The control box shown in place, Fig. 156, front view, and Fig. 157, back view, with covers removed, contains an eccentric and a cam for each cylinder keyed to the shaft, at the end of which is the governor. Fig. 158 shows end of engine with control box removed, showing shaft and governor, and with other parts completely exposed. Each eccentric drives a fuel injection pump plunger push rod. Each cam operates an air starting valve until the engine is brought up to speed, after which the cam automatically goes out of action, with the shutting off of air at the master valve, until another start is to be made.

There are no moving parts outside of the control box except the linkages connecting the speed control governor to the injection pump rocker shaft, which pass through the control box casing at two points, so that the operator can assure himself by watching their movement that these parts are free and the engine under governor control.

Cover plates are provided on all sides oı the control box to permit of inspection, but are normally bolted tight to keep lubricating oil in, and dust or moisture of the air out.

The eccentric cams and governor dip in oil in the bottom of the control box, fed by the end bearing throw-off and overflowing to the crank pit sump. This insures perfect automatic lubrication of these parts.

There are two practically independent parts to the systems for handling the fuel oil from the main storage tank to the injection chamber. The first circulates fuel oil from the storage tank, usually underground and outside the building, to a suction box on the injection pump, the overflow or excess flowing back to the storage tank. The second takes the oil from the suction box through the fuel injection pump and delivers it to the sprayer at the top of each stroke in quantities metered by the governor to suit the load.

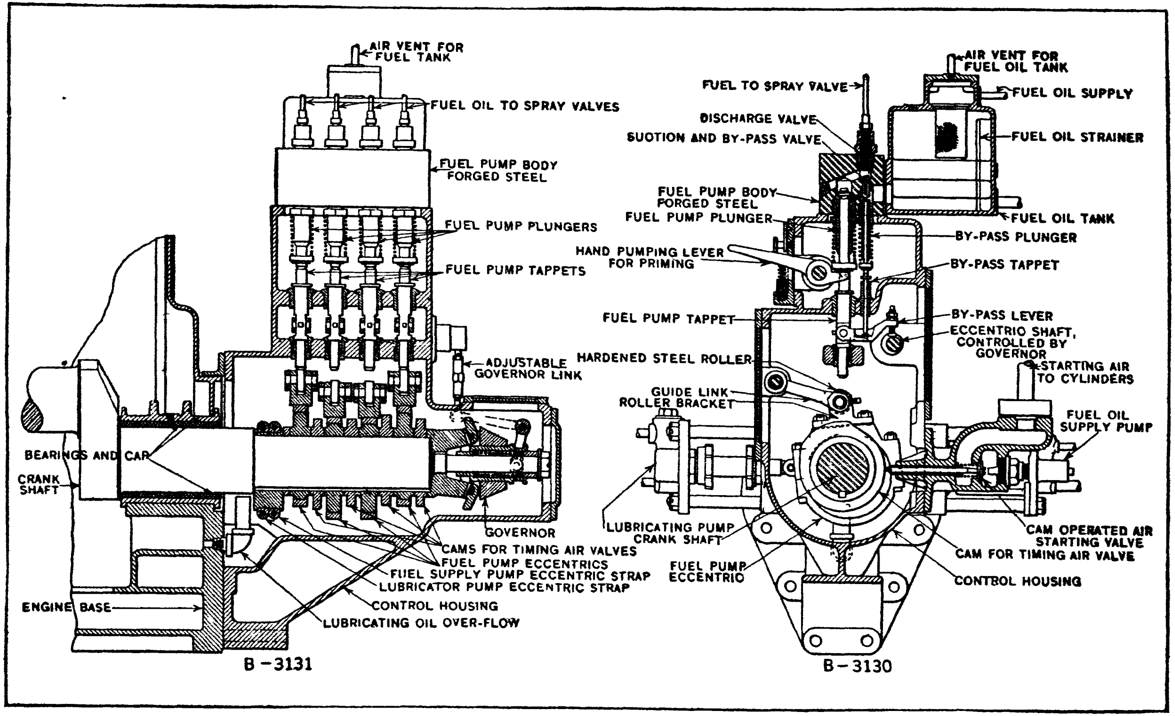


Fig. 159.—Control Box Sectional Assemblies, Air Starting, Fuel Service and Injection Pumps, Lubricating Oil Pump and Governor, Worthington Engine

Located on the top of the control box with its driving and control mechanism within the control box, the fuel pump is shown in Fig. 159 in sectional assembly for a four cylinder engine. Inside the control box there is an eccentric for each cylinder keyed to the crankshaft with air starting cams between. At the end of the shaft is the governor, and close to the engine base the eccentric for driving the fuel oil and the main lubricating oil circulating pumps for those engines that do not use oil-cooled pistons. In cases where oil-cooled pistons are used, a gear pump is substituted for lubricating and piston cooling oil. These fuel pump eccentrics are each guided at the top by a side thrust rod carrying a contact roller on the pin joint. This contact roller strikes the lower end of the fuel pump plunger push rod. The parts are all splash lubricated, as are also the other parts in the control box below the horizontal partition separating the lubricating system from the fuel oil system. These parts include the horizontal lever, pin connected to the plunger push rod, the lower end of the bypass push rod, and the rock shaft actuated by the governor, and carrying the eccentric fulcrums of the fixed end of the horizontal lever.

The pump body proper is machined from a forged steel billet to absolutely prevent porosity and to insure a soundness of metal for oil under the pressure produced at each injection. Holes are bored to receive the discharge valve assembly on the top, the large suction port at the right-hand side, the bypass push rod assembly on the bottom at the right and the pump plunger assembly on the bottom, at the left.

The pump plunger is so accurately fitted in its removable sleeve, or working barrel, as to work freely without oil leaks and as no stuffing box is used, trouble with packing, often so common, is here rendered impossible.

A hand lever plunger lifter for each cylinder at the side, see Fig. 159, makes it possible to prime the spray nozzle and its pipe connection from the pump discharge valve for removing air before starting the engine. This plunger lifter may also be used for stopping the engine; the plunger being raised, oil injection ceases.

From the suction box the fuel oil is drawn into the pump

by the down strokes of the plunger through the suction valve. This suction down stroke of the plunger is spring actuated, the spring causing the plunger to follow an eccentric till it strikes a stop. The plunger then remains stationary with the pump chamber full of oil for nearly a whole revolution, and until it

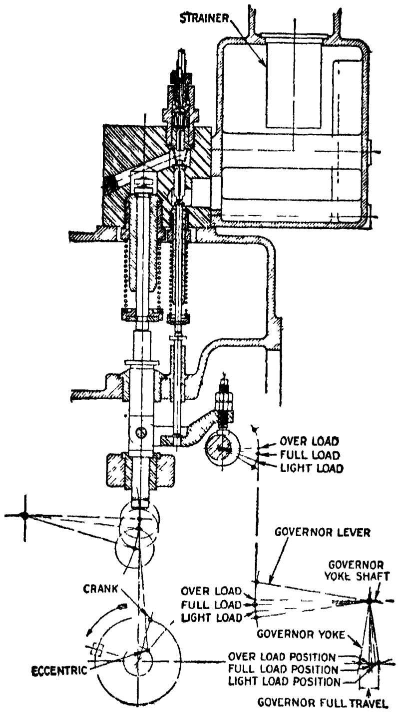


Fig. 160.—Fuel Injection Pump Section and Timing Diagram with Governor Control, Worthington Engine

is driven up by the eccentric keyed to the crankshaft. This contact occurs when the crank is 16 degrees from the cylinder axis, at which time the line of eccentricity is 25 degrees from the crank, or 41 degrees from the cylinder axis. This position is the beginning of injection at all times, see Fig. 160.

Injection always ends before the plunger has completed its full stroke. At full load this occurs when the plunger has completed about half of its stroke and sooner at lighter

loads. Delivery ends when the suction valve is mechanically opened by the plunger acting through a horizontal lever, one end of which is lifted by the plunger, the other end being fixed. The middle of this lever raises the suction valve by a push rod, and the time when this happens depends on the height of the fixed end, or fulcrum, of the horizontal lever.

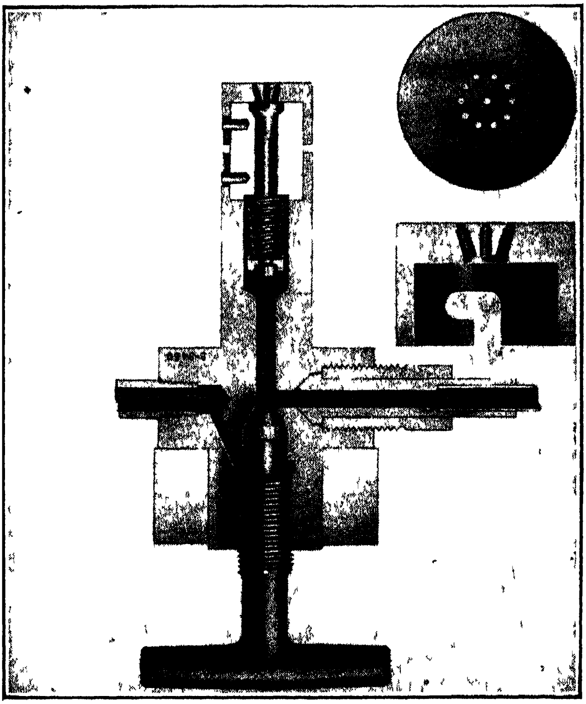


Fig. 161.—Fuel Sprayer, Worthington Engine

This in turn is adjusted automatically by the governor, which rotates the eccentric that is the fulcrum, as shown in Fig. 160.

From the pump delivery valve the fuel oil passes through a seamless steel tube with rigid walls, so it cannot expand appreciably by the oil pressure, to the sprayer on the cylinder head. This sprayer is of simple construction as shown in Fig. 161. Essentially it has a forged tube body, the lower end of which carries a bayonet joint cap which is the sprayer

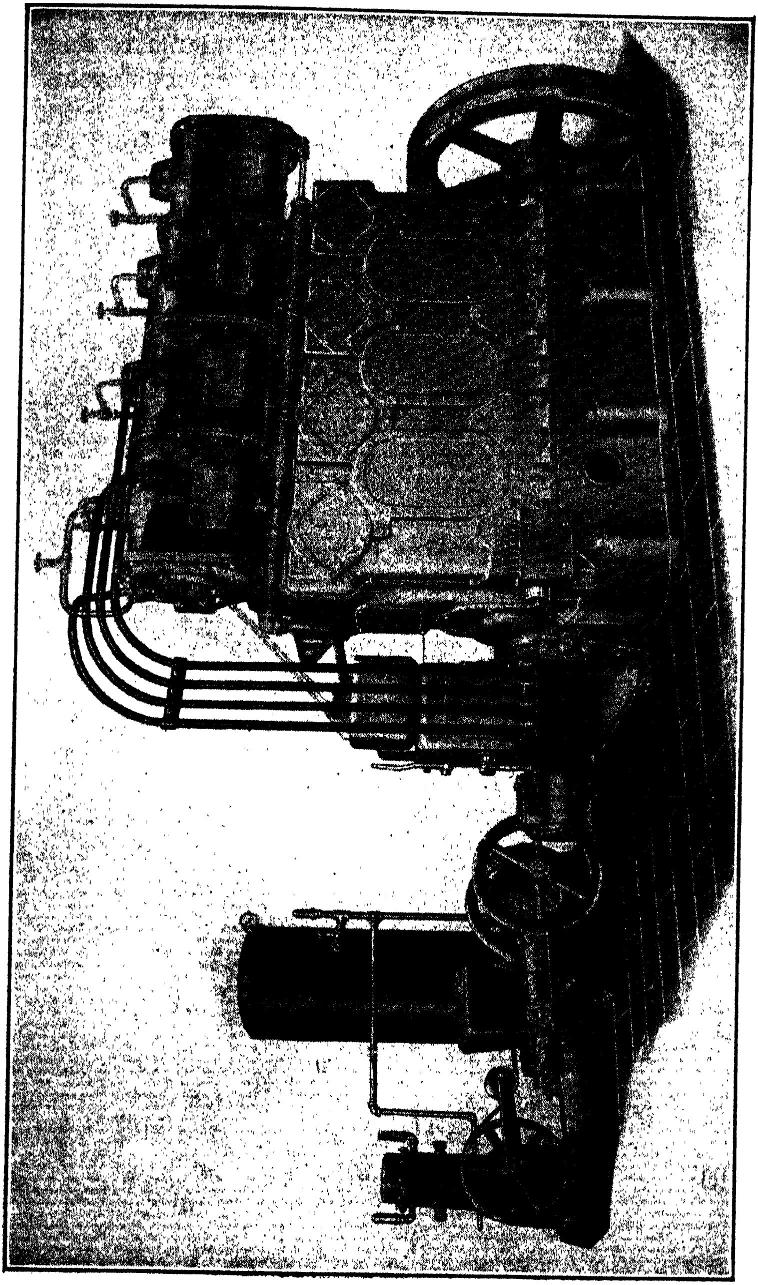


Fig. 162.—Air Starting System, Worthington Engine

proper. The outer face or spray plate carries the ten holes in a circle slightly diverging from one in the middle to deliver the oil across the injection chamber and finally deposit it around the ejection orifice ring ready to enter the cylinder when at the right time the pressures start a flow from injection chamber to cylinder. A check valve behind the spray plate prevents back flow of cylinder gases and keeps the oil pipe full of oil. At the top of the body there is a hand vent or bleeder to be opened before starting the engine, so that by operating the hand lever of each pump plunger, the oil pipe and sprayer body can be filled with oil. The open vent valve showing a clean spurt at each stroke without air bubbles, is then closed and remains closed.

On the extreme end of the crankshaft the governor is keyed, within the control box for automatic lubrication and dust proofing, but its linkage connection to the pump control rock shaft is carried outside. This outside linkage arrangement is for the convenience of the operator as a means of assuring him at all times that these parts, vital to the safety of the engine, are free and in good working order. A pair of balanced weights pivoted to the governor body proper have bell cranks that press against a sleeve opposed by the springs around the governor spindle held by the end nut. As speed rises, the sleeve compresses the spring as it moves toward the shaft end, and carries a lever hanging from the pin that passes through the control box. In service any speed change rotates this pin through a very small angle, and by the outside linkage the pump bypass control rock shaft is also rotated through a small angle, adjusting pump delivery of all pumps equally to restore speed to normal.

Between each pump delivery the rock shaft carrying the bypass lever fulcrum is absolutely free, so that its adjustment by the governor requires practically no force at all. This insures most accurate speed control with a quite small governor. A factor that contributes to close regulation is the fact that fuel is controlled to load requirements at the beginning of each working stroke; that is, every down stroke, and there is no time lag between the governor movement and the effect on power adjustment, as there is in all four-cycle engines.

Compressed air is used for starting the engine in all cases, and the control of the air to act as motive fluid in the working cylinders is entirely automatic, but the pumping up of new supplies of air in the air storage tanks, after each start, is a hand control operation. The system is a double one, involving

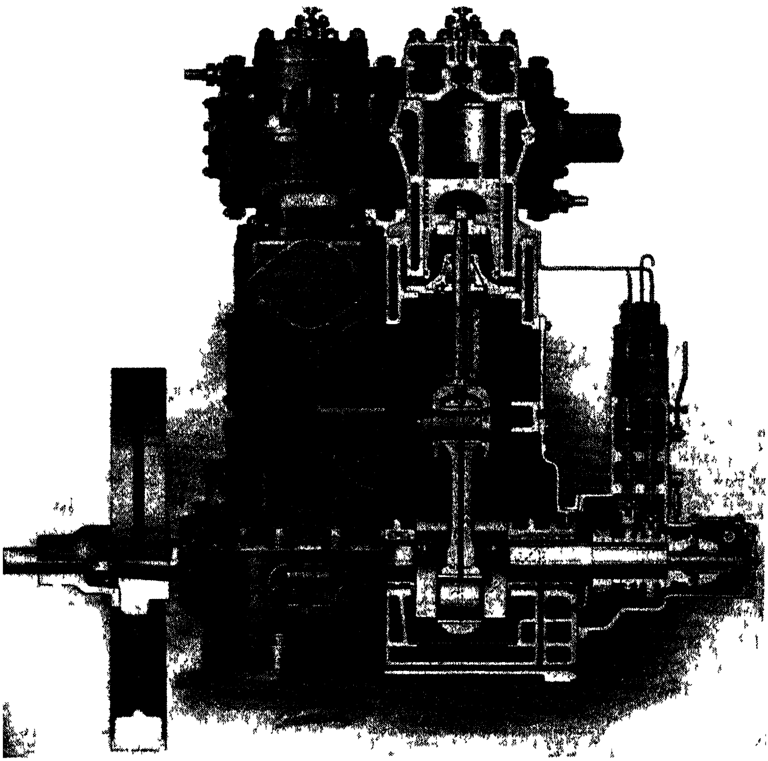


Fig. 163.—Longitudinal Section of Worthington Engine Showing Pistons Indirectly Cooled

an air supply part and an air distributing part, the two parts joining at the hand control air stop valve.

Compressed air at the full pressure is brought to the engine at the hand air valve. Whenever this lever is lifted by hand, compressed air becomes available at the mechanically operated air distributing valves on the control box. These valves per-

mit air to flow to any cylinder in which the piston is moving down and cuts off before the down stroke is completed. Air thus mechanically distributed to the several cylinders enters

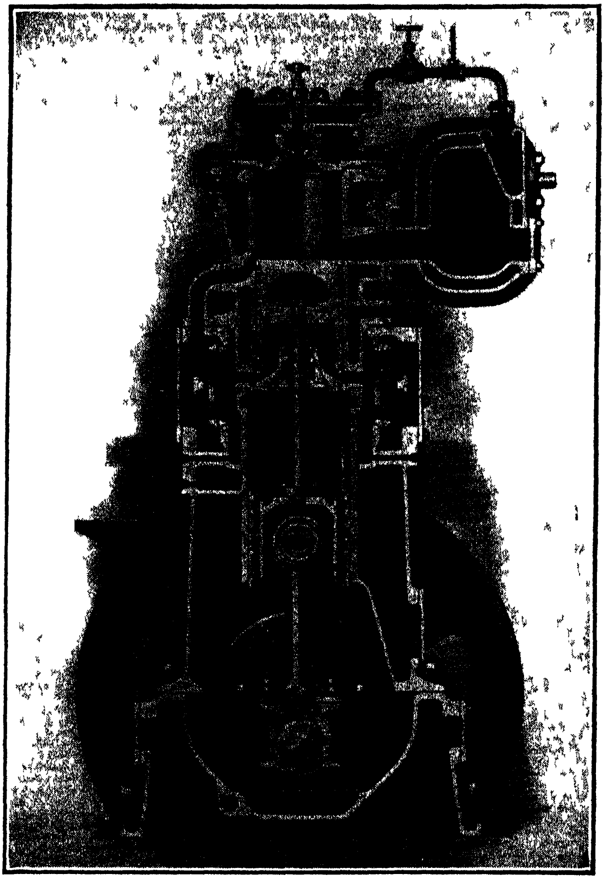


Fig. 164.—Transverse Section of Worthington Engine Showing Pistons Indirectly Cooled

them through check valves in the cylinder head. The construction of the compressed air distributing valves and the operating cams is shown in Fig. 159, the cylinder head check valves in Fig. 150, and the whole system in Fig. 162.

When compressed air reaches the distributing valves, after lifting of the lever of the hand air valve, the distributing valves are forced to their seats, and stems are brought within range of the cams located beside each pump eccentric in the control box. As these cams rotate with the shaft, the air valves are opened and closed at the right time to supply compressed air to the cylinders for each down stroke, so long as the hand lever is held up. Just as soon as the hand lever is dropped, no more air can reach the distributing valve, springs lift them off their seats, and the stem ends move out of cam contact, so no wear takes place as long as the engine is in operation.

The actual operation of starting the engine is very simple. As a first step the oil pipes are filled by the hand pump until oil shows at the sprayer vent on top of the cylinder head. By hand bar, the flywheel is turned till one piston is ready to start down, except for four or more cylinder engines which always have a piston in this position. Raising the hand air valve lever will then start the engine on compressed air. After a few turns, fuel oil delivered by the pumps is fired regularly, and the compressed air lever dropped. The engine may be loaded immediately.

As will be noted from the foregoing description of engine details, two methods have been adopted for limiting piston and cylinder temperature differences. In engines on which motive cylinders of thirty horsepower occur, these differences are controlled by a proper disposition and thickness of metal in the design of the parts supplemented by a suitable system of water jackets. In engines of larger size, those with fifty and seventy-five horsepower cylinders, this limit of temperature difference is further insured by means of the complete oil cooling system applied to the piston.

A study of sectional views, Fig. 163 and 164, illustrating the arrangement of parts on the smaller size engines, and Figs. 152 and 165, covering the construction of the larger sized engines, will show these differences in detail.

The Worthington 2-cycle solid injection marine engines are of similar design to the stationary engines, except for a flywheel of smaller diameter with wider face to swing in a

pit inside the sub-base. For marine use the following modifications of detail have been specially developed as elements of difference over the stationary engine previously described.

(a) Reversing, variable speed injection pump and air starting mechanism in control box with hand levers.

(b) Elongated bed plate with holding-down flanges near shaft center level to match floor plates and with pump assembly, flywheel and thrust bearing carried on the after extension.

(c) Water jacket circulating, bilge, fuel oil service and forced feed lubricating oil pumps in one assembly, driven from a single eccentric on the main shaft, with two stage starting or manœuvering air compressor, all automatically lubricated.

(d) Flywheel of reduced diameter swinging inside bed plate.

(e) Thrust bearing at after end of bed plate extension.

A special feature is the carrying of the lubricating oil sump beyond the engine to a partition in the bed plate extension so that the attached pump assembly shown in section in Fig. 166, is force feed lubricated from the main engine system. On the extension of the crankshaft, there is mounted an eccentric beyond the last main bearing as shown in this drawing. This eccentric drives all the attached pumps and a two stage compressor for charging the manœuvering air tanks. The number and size of these tanks are determined by the frequency of manœuvers typical of the service of the vessel.

Provision is made for four attached pumps. The lower pumps are for bilge and cooling water. The upper pumps are for fuel oil service and lubricating oil.

A housing encloses the whole pump and compressor assembly.

Without departing from the principles of injection pump operation and timing and of starting air valve timing used in the stationary engines, these are modified slightly for the reversing marine engine so it can start in either direction at the will of the operator manipulating an air start or reverse

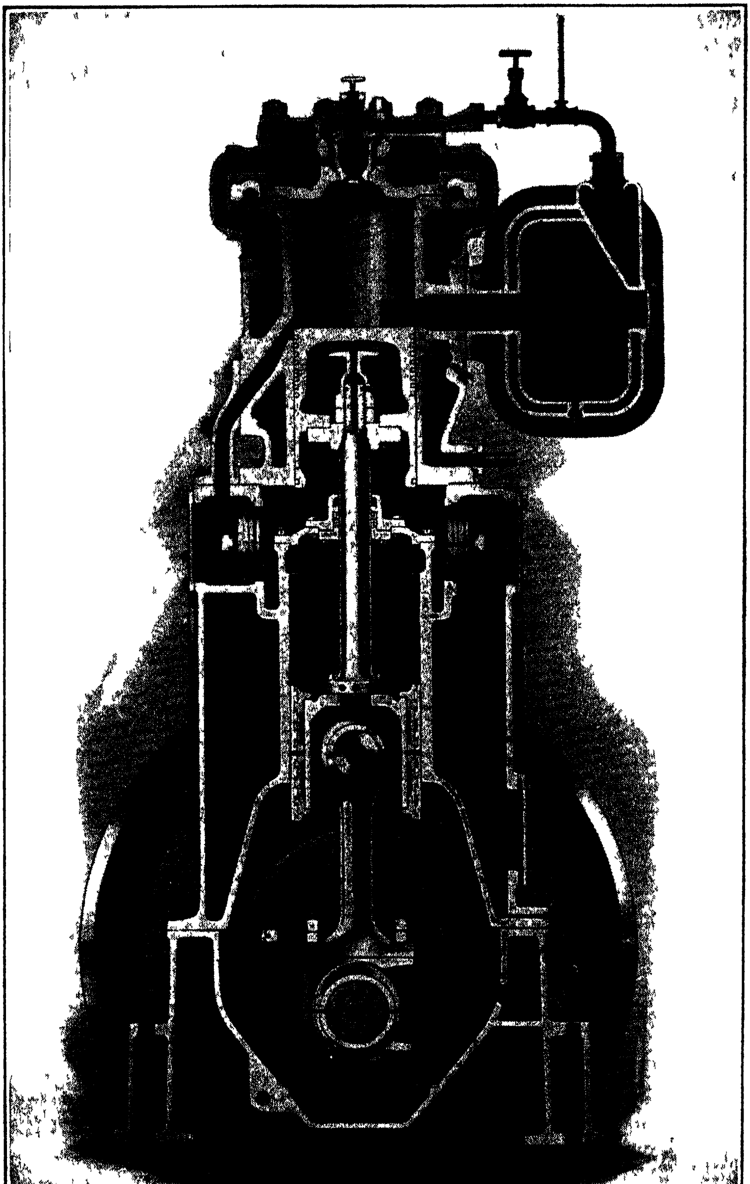


Fig. 165.—Transverse Section of Worthington Engine Showing Arrangement of Scavenging Air and Directly Oil Cooled Pistons

lever, and the engine run at any speed by setting a second or fuel lever.

The eccentrics that drive fuel oil injection pumps are

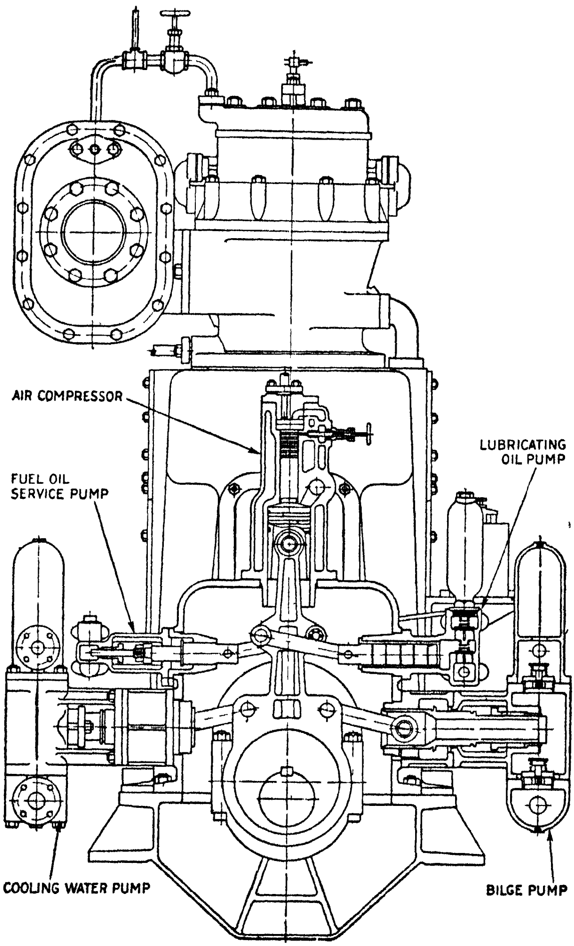


Fig. 166.—Attached Pumps and Manoeuvering Air Compressor, Worthington Marine Engine

machined in a sleeve that slides freely on the crankshaft between the governor and a collar next to the main bearing in the control box. A lug in this sleeve works in a radially

slotted sector of the collar and the collar drives the sleeve by contact of the lug at one or the other end of the slot, depending on the way the engine was started. This is shown in detail in Fig. 167, with the air cams, which in this case are carried on a cam shaft, gear driven inside the control box. Except at the moment of starting these parts are free and have nothing to do.

There are eight air cams and four air valves for a four cylinder engine, two for each cylinder on a collar sliding on the shaft, one set fixed for ahead and the other set for reverse,

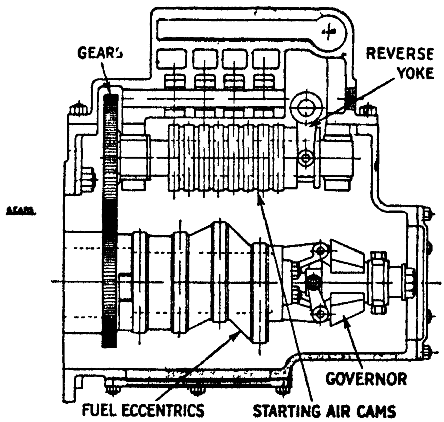


Fig. 167.—Reversing Fuel Eccentrics and Starting Air Cams, Worthington Marine Engine

and longitudinal movement of the sleeve brings one set or the other into line with the valve stem rollers.

The whole assembly of the reversing mechanism in the control box with the two hand control levers is shown in side and end view in Fig. 168.

The actual reversing of the engine is done by the air reversing lever, which performs two functions as it is moved either way from mid or neutral positions:

1. To shift the set of air cams.
2. To open the master air valve.

The shifting of the air cam by means of the air reversing lever will bring either the ahead or astern cams into action

on the air valves, thus imparting to the engine the desired direction of rotation. There is an interlock between the speed control and the reversing levers which will not permit the reversing lever to be thrown to either of its extreme positions until the fuel control lever is brought to its "no fuel" position. After the air reversing lever is thrown into the desired position for either ahead or astern rotation, the fuel control lever

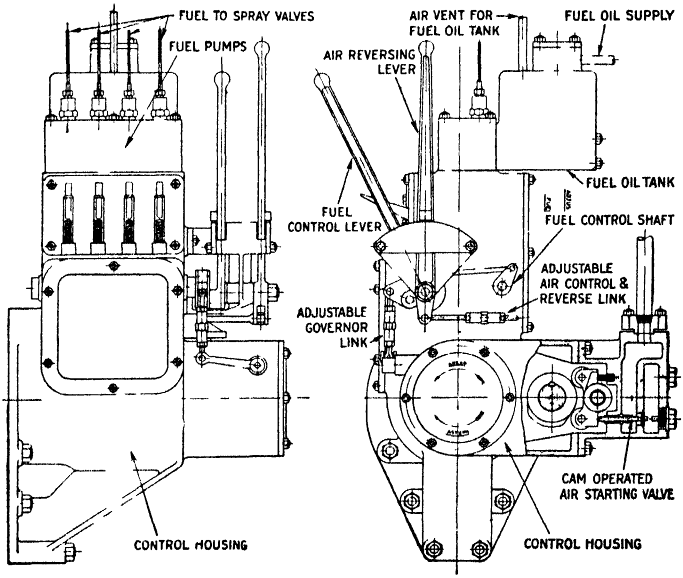


Fig. 168—**Reversing Gear and Controls, Worthington Marine Engine**

is then moved to the required "fuel" position, whereupon the air reversing lever is moved on to air starting position. The engine now starts and immediately begins firing and runs at the desired speed. The air reversing lever is then brought back to its running position, shutting off the starting air. The fuel control lever carries a bell crank lever at its lower end, and the required fuel control is attained by a change in the location of the fulcrum of this bell crank lever, one arm of which is connected to the governor links and the other to the control shaft. Movement of the fuel control levers necessarily shifts the fulcrum of the aforementioned bell crank

lever, thereby effecting the necessary fuel control by means of its action on the control shaft. The governor is always in control of maximum speed.

McIntosh & Seymour Diesel Engines. This firm has the largest factory solely devoted to the manufacture of Diesel engines in this country. Their range of sizes is from 80 to 8,000 H.P. and they build both marine and stationary types of engines on the four-cycle principle.

Fig. 169 shows very clearly the general design of the base for the trunk piston type McIntosh & Seymour Stationary

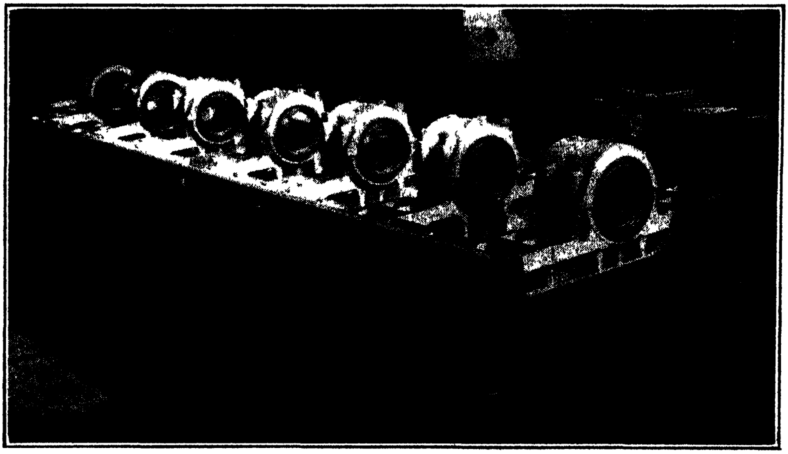


Fig. 169.—Base Plate, McIntosh and Seymour Engine

engine. The bolting for attaching the frame is as close as possible to the bearing, leaving convenient access to the main bearing cap. This makes a minimum length of bearing girder and insures great stiffness as well as great strength with a minimum weight. The ends of the bearing girders are carried by longitudinal members, which together with the solid bottom in the base makes a base of great longitudinal strength and stiffness.

The main bearing seatings are carefully machined at one setting insuring perfect uniformity and alignment, making long life for the running parts possible.

The side walls of the frame extend from the outside of the cylinder flanges directly to the bolting on the base, carrying the principal stresses in a very direct manner. Two transverse ribs connect the sides of the frame between the cylinders and are extended through to the outside of the frame forming the opening to give the necessary clearance for the cranks and to give convenient access to them. This

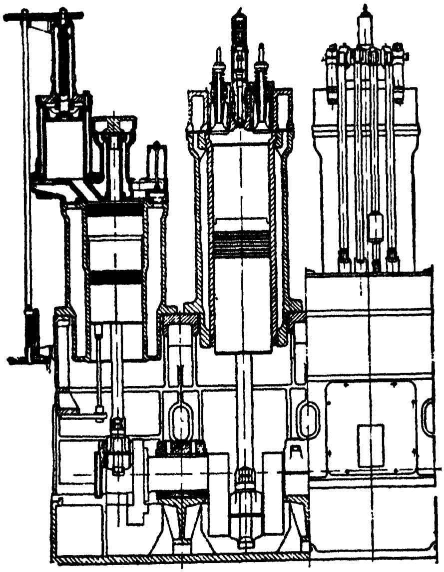


Fig. 170.—Sectional View of Working Cylinder and Compressor, McIntosh and Seymour Trunk Piston Type Engine

arrangement gives a frame of great stiffness, both for the direct and transverse stresses, with a minimum weight. Handholes are provided opposite each bearing. Substantial sheet metal doors are provided over the crank openings with sliding doors to give convenient access and inspection while the engine is in operation without chance for throwing of lubricating oil on the floor. These doors are narrower than the crankpin boxes, in this way preventing the oil from the ends of the crankpin bearings being thrown out.

The compressor is attached to the forward end of the engine and is of the three stage type. Cooling is accomplished

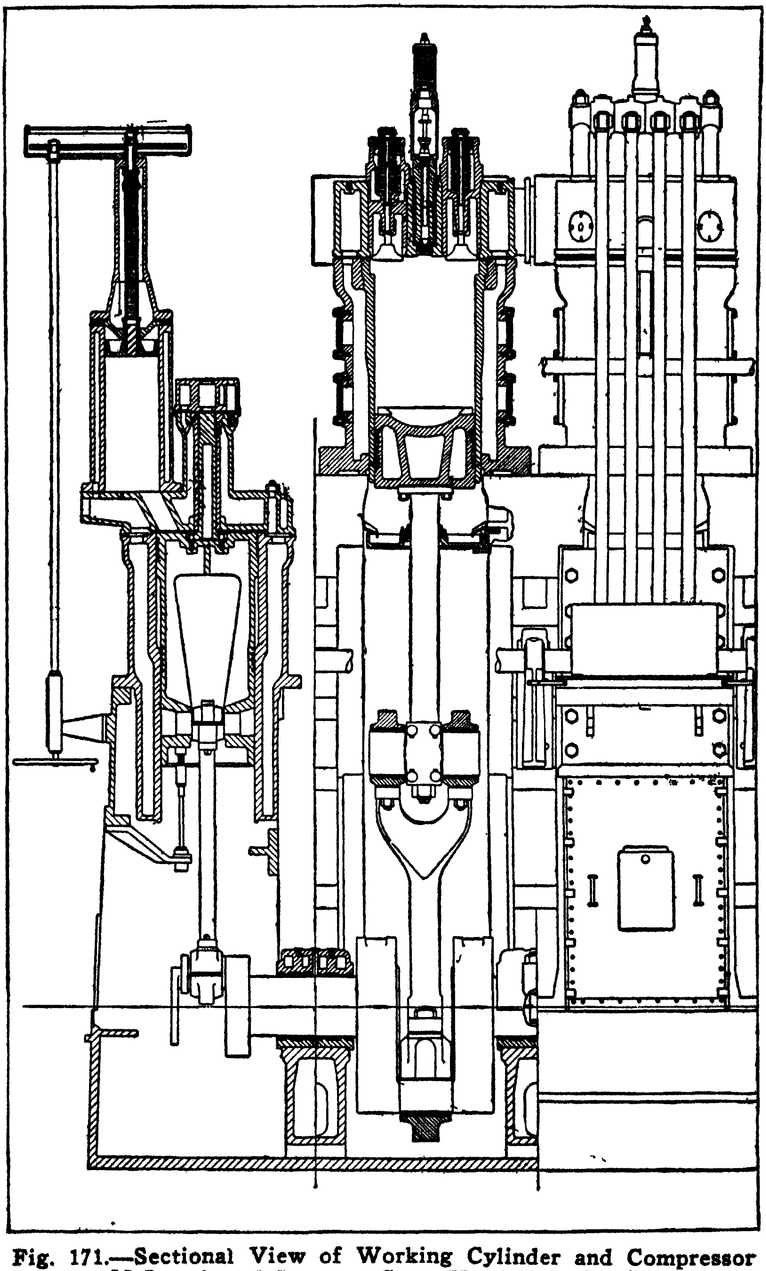


Fig. 171.—Sectional View of Working Cylinder and Compressor McIntosh and Seymour Cross-Head Type Engine

after each stage. A section through the air compressor for the trunk piston type engines is shown in Fig. No. 170 while that for the cross head type engines is shown in Fig. No. 171.

The crankshafts for the small size engines are solid forgings, while those for the larger sizes are built up in sections and the sections bolted together. An overhung crank is fastened to one end for driving the air compressor, and the extension shaft is coupled to the other end of the shaft.

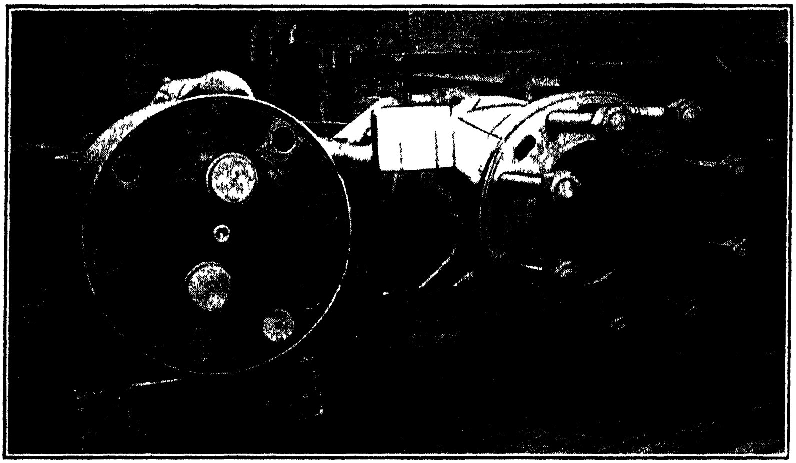


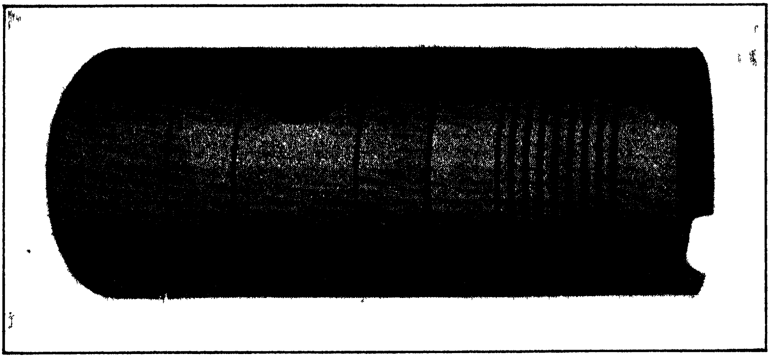
Fig. 172.—Working Cylinder and Cylinder Head, McIntosh and Seymour Engine

All the shafts are accurately machined, are carefully inspected, and held within very close limits. During the manufacture these shafts are carefully annealed at least twice, test pieces are taken from between the webs, and every care is taken to insure perfect shafts.

As shown by the illustration Fig. 172 the cylinder is made in two separate parts. A substantial jacket carries the main stresses, forms the outer wall of the water space, and supports the renewable liner. These liners, which are of a special mixture of charcoal iron to give the best wearing qualities, are secured at the upper end and free to expand at the lower end.

Openings between the cylinder studs give thorough cooling at the hottest part of the cylinder.

The McIntosh & Seymour trunk pistons Fig. 173 are cast



No. 173.—Piston, McIntosh and Seymour Trunk Piston Type Engine

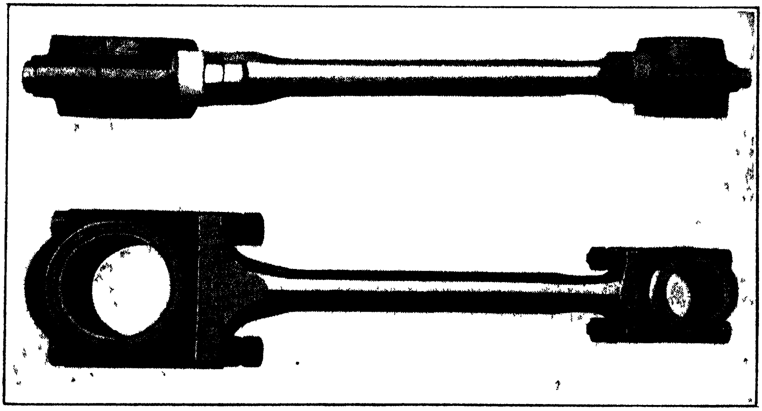


Fig. 174.—Connecting Rod of McIntosh and Seymour Trunk Piston Type Engine

in dry sand, of a special charcoal iron mixture with ample riser and receive a special heat treatment before machining.

The piston is first accurately machined, the piston pin hole being bored in a special fixture making the pin exactly at right angles to the axis of the piston, and finally the piston

is ground to the size and shape that has proved to give the best possible service in the engine.

The connecting rod Fig. 174 is of the marine type with steel boxes lined with a very hard genuine tin base babbitt. The rods and boxes are provided with laminated liners for adjustment.

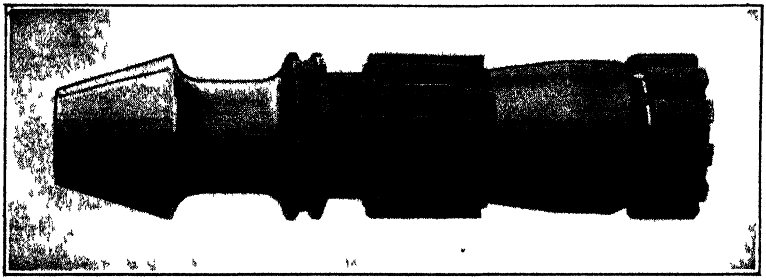


Fig. 175.—McIntosh and Seymour Atomizer

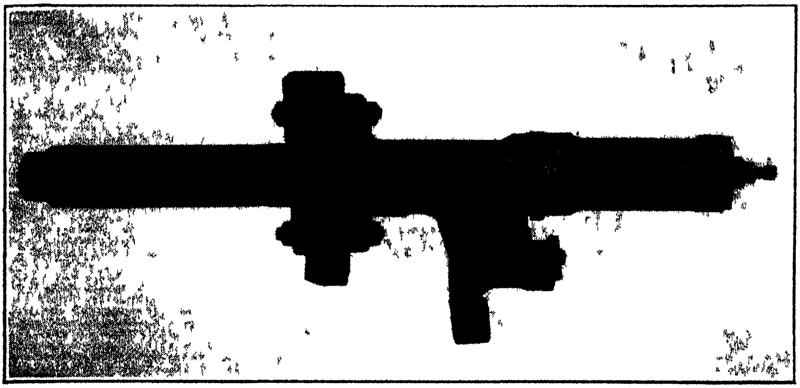


Fig. 176.—McIntosh and Seymour Fuel Valve

The piston for the crosshead engine is of box form, cast in dry sand, of a special charcoal iron mixture, is accurately machined and provided with an ample number of cast iron packing rings, and are arranged for water cooling.

The exhaust valves are of cast iron fused to steel stems, working in interchangeable cages with a spring of great

length to give maximum life. The valve tappet connects to the upper end of the stem.

The atomizer shown in Fig. 175 is the most important part of the Diesel engine as it is the means of the preparation of the fuel for the burning, and on the effectiveness of this preparation depend not only the fuel economy, but what is much more important, the heat conditions in the cylinder.

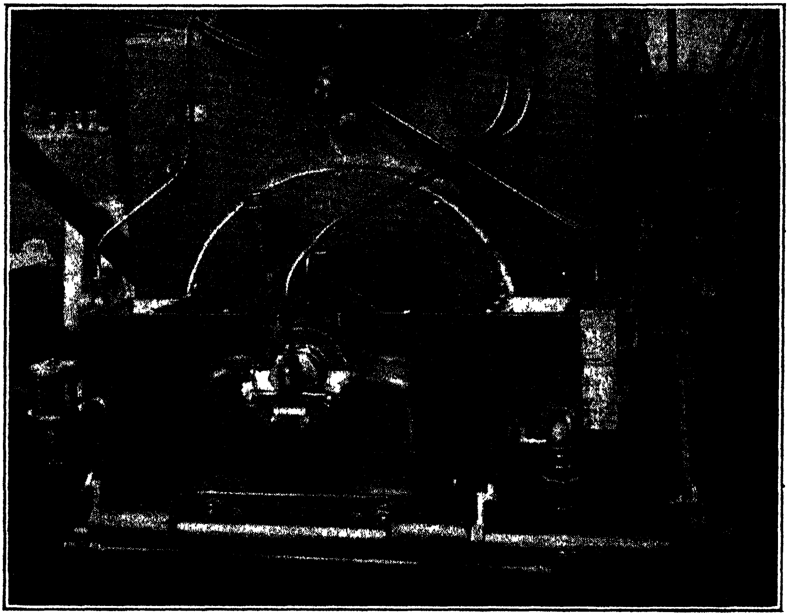


Fig. 177.—McIntosh and Seymour Fuel Pumps

The atomizer fits snugly inside the lower fuel valve body, the tapered part at the bottom being forced against a corresponding seat making a tight joint. The fuel valve needle passes down through the inside of the pulverizer, being a sliding fit at the upper end but with a gradually increasing clearance toward the lower end. The fuel stands at the level of the delivery holes at the bottom of the slots in the middle of the atomizer, and while the needle valve is closed the fuel pump delivers a charge of oil into the annular space around the outside of the lower part of the atomizer. The oil rises uni-

formly through the slotted diaphragms shown into the slots above the discharge openings.

When the needle valve opens the atomizing air passes down through the slots and holes at the top of the atomizer and the unbalanced pressure created forces the fuel oil up

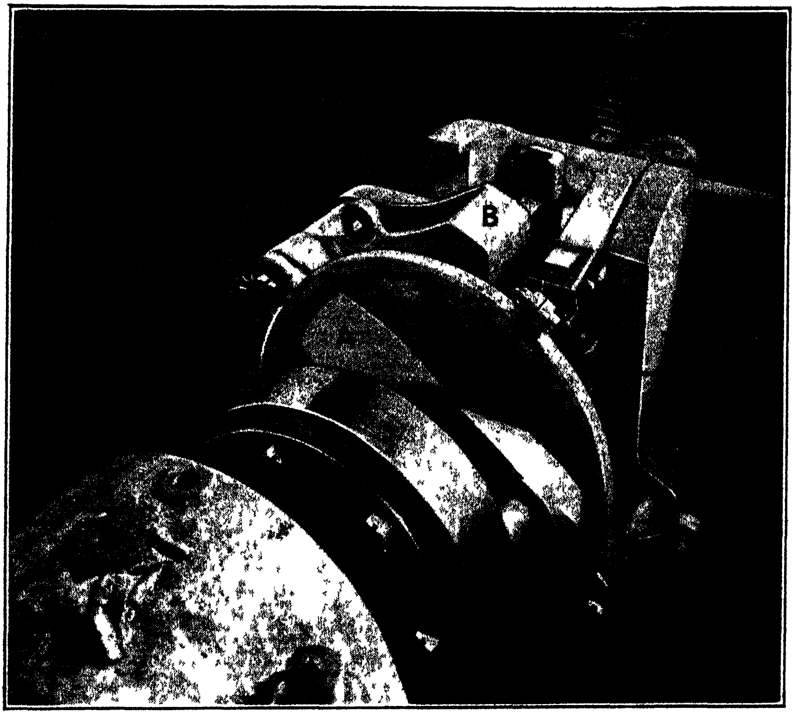


Fig. 178.—McIntosh and Seymour Governor

through the delivery holes shown and is very effectively broken up into a very fine mist by the swiftly moving air passing down by the needle.

The fuel valve, Fig. 176, is of cast iron, contains in the lower body the atomizer, in the upper body the spring which closes the needle valve, and in the middle the actuating mechanism for opening the needle valve. At the lower end is attached the distributing plate which throws the atomized oil

out in a cone-shaped jet so as to get the fuel in close contact with the heated air and give the turbulence to the burning mixture so absolutely necessary to complete combustion.

Fig. 177 shows a separate fuel pump for each cylinder, the plungers for all of these pumps are driven from one eccentric controlled by the governor which at all times maintains the proper stroke to deliver to each cylinder just the amount of oil necessary to carry the load on the engine.

The fuel oil is supplied to the fuel pumps under a slight head from the daily service tank in the station through the pipe shown under the pumps and each pump delivers the fuel oil through double delivery valves to the small pipes leading to the fuel valve on each cylinder.

The mechanism for control shown underneath the pumps is arranged to lift the suction valves to stop the engine, or to cut out a single pump, if so desired.

Fig. 178 shows very clearly how the governor is arranged at the compressor end of the crankshaft.

The spider "A" carries the two governor weights—one of which is shown at "B". One of the weights carries a small block which slides in a groove in an extension of the swinging eccentric so that when the weights are in their outer position, this swinging eccentric runs central, and the plunger of the fuel pump can deliver no oil. When the weights are in their inner position the eccentric has its maximum eccentricity and the plungers have sufficient stroke to deliver the maximum amount of oil required. The centrifugal force of these weights is balanced at rated speed by the springs "C" so that if the speed increases the weights move out enough to reduce the amount of oil pumped to just that necessary to carry the load and when more load comes on the engine tends to slow down and the weights move in and increase the amount of fuel oil pumped to just the proper amount.

It can very easily be seen that the weights act very efficiently through this wedging action on the eccentric but the reaction of the eccentric comes practically within the angle of friction so as to have practically no effect on the weights.

The certainty of action of this governor makes it absolutely reliable as it depends on no relay devices whatever, and

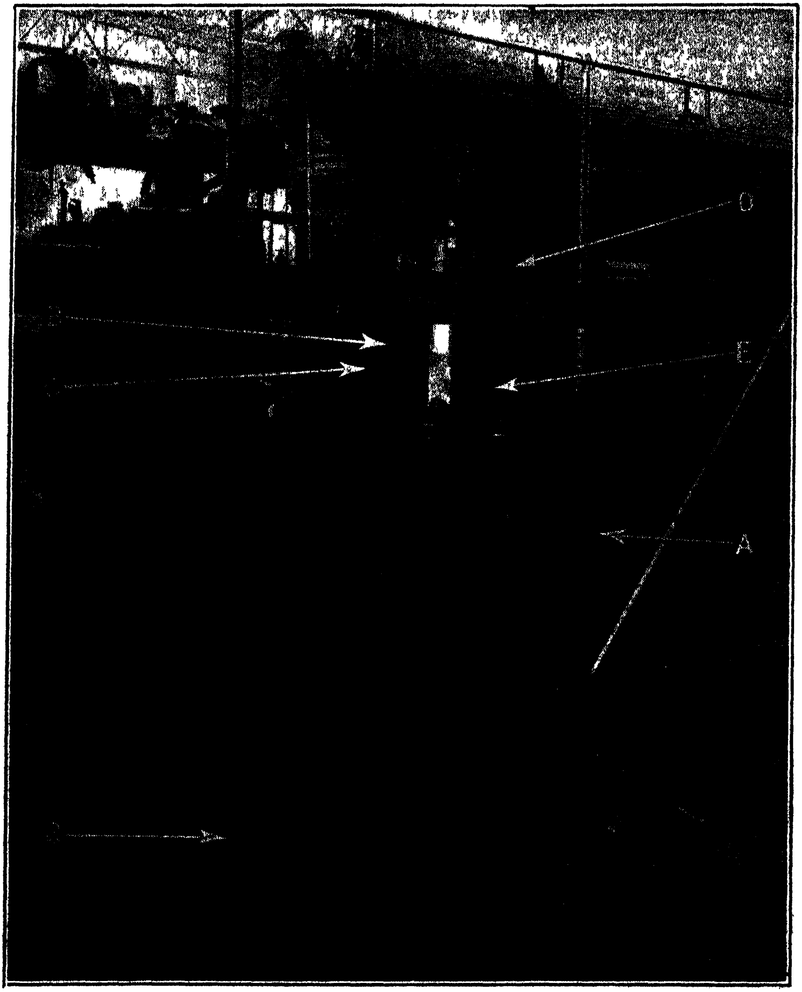


Fig. 179.—Lubricating Arrangement of McIntosh and Seymour Trunk Piston Type Engine

has the whole power of the engine behind it, to make it function.

The fundamental principles of the lubrication of Diesel engines is to lubricate the pistons without any excess of oil, therefore, on all McIntosh & Seymour standard trunk piston type engines the general lubricating arrangements are as

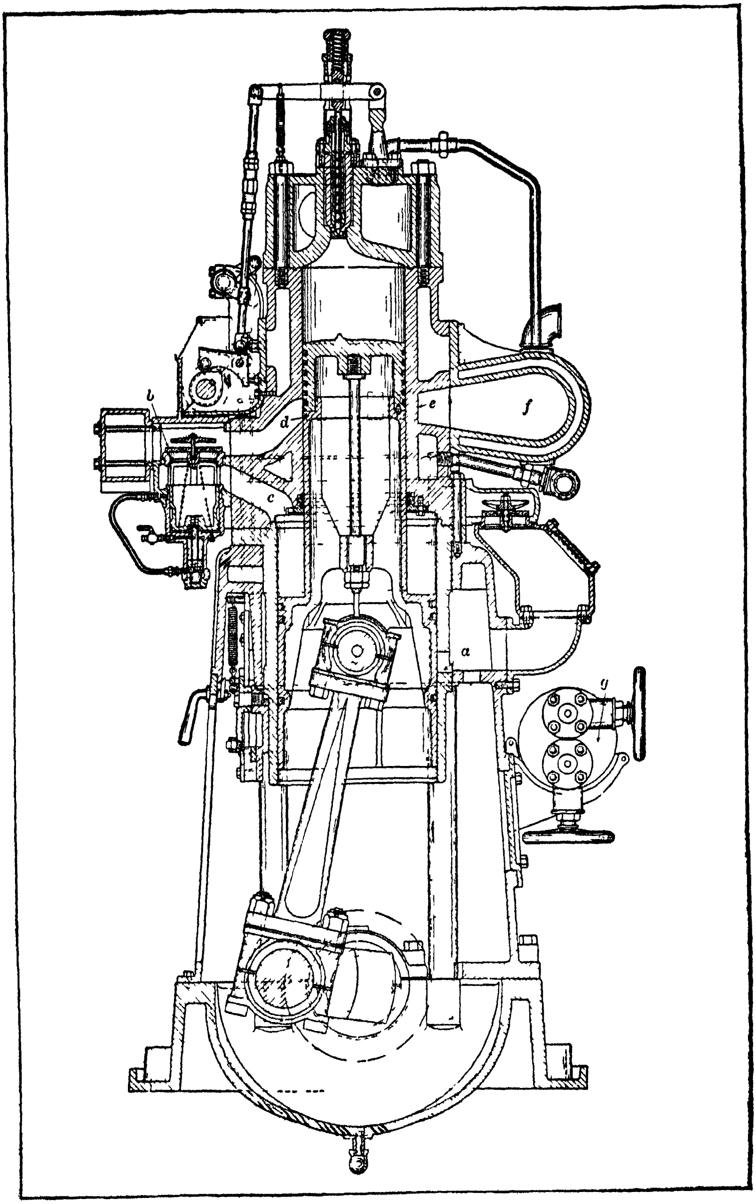


Fig. 180.—Cross Section Through Southwark-Harris Engine

shown by Fig. 179, and it consists of the force feed lubricator "A" which supplies oil by individual plungers to the main pistons through the cylinder wall at two places 180° apart, as at "B", to the main piston pins through the cylinder wall and into a slot in the piston as at "C" and to the compressor cylinders. The main bearings and crank pins are lubricated by gravity pressures from the gang oilers "D", there being two feeds to each main bearing and one to each crank pin, the oil being delivered through pipes "E".

The valve levers are arranged for hand lubrication. This insures regular periods for inspection of these parts by the operator.

The engine base has a closed bottom so that all oil fed to the working parts is caught and flows to a sump tank and the plunger pump "G" returns all the oil to a filter where it is settled and filtered and then goes to a small overhead tank where it again flows to the oilers on the engine. This arrangement of re-use of the oil insures continuous lubrication with a minimum lubricating oil consumption.

Southwark-Harris Marine Diesel Engines. These engines are of the 2-cycle, step-piston type, each working cylinder having its own scavenging pump. The scavenging pistons are an enlarged extension of the working piston and work in their own cylinders.

The scavenging piston, besides supplying low pressure air for scavenging the working cylinders, plays an important part when starting the engine by means of compressed air. Upon the movement of the starting lever either "ahead" or "astern", each scavenging cylinder becomes immediately converted into an air motor by the automatic cutting off of the suction and discharge valves of the scavenging pump. By the continued movement of the handling gear the spray valves come into operation and fuel commences to be supplied to the working cylinders, the engine still running on compressed air in the scavenging cylinders without affecting the working conditions in the main cylinders, thereby avoiding the admission of the usual high pressure air into the working cylinders, just at the time it is necessary to build up the working temperature. This renders it an easy and quick engine to maneuver, also assur-

ing certainty, as it is not necessary to shut off the fuel, nor interfere with conditions in the working cylinders in any way, at the time of reversing. In starting and reversing this engine the spray valves do not operate while starting on air until the operator wishes them to do so, consequently all the usual by-pass valves are eliminated and the temperature of the fresh charge of air in the cylinder is quickly raised, owing to the absence of the chilling effect caused by the blast air being allowed to enter through the spray valve without fuel.

The engine has no valves in its cylinder heads and they have only to withstand the normal working pressure. The method of using the scavenging piston for air starting does away with the troubles and necessity of air starting valves in the cylinder heads.

The scavenging air is admitted to the cylinders through ports in their circumference. The exhaust gases pass out through ports located opposite the scavenging ports and are so arranged that the piston opens and shuts them at the correct time during its travel.

There is one fuel pump for each cylinder of the variable stroke type, this variation being controlled automatically by the governor. The governor controls the discharge and not the suction valves of the fuel pump and this does away with the necessity of by-passing, and the oil is not churned as sometimes happens with engines having the governor control on the suction valves. A cross sectional view of the Southwark-Harris engine is shown in Fig. 180.

Busch-Sulzer Bros. 2-Cycle Marine Engines. These engines are of the vertical, four or six cylinder, single acting, cross-head type. The 1800 B.H.P. engine is shown in Fig. 181 and a cross-sectional view of their stationary engine, which is of the same general construction as the marine type, is shown in Fig. 182.

The bed plate is built up in sections, of strong, medium-soft cast iron. The crank case is oil and gas tight, and of the enclosed type. It is built up of sections and is rigidly bolted to the top of the bedplate and provided with large covers, readily removable for inspection and adjustments, making all parts inside the crank case easily accessible. The covers carry

hinged inspection doors, for convenience in making inspection while the engine is in operation, without throwing oil. The crank case carries the cross-head guides, and the cylinder jackets are bolted directly to its top.

The working cylinder consists of two main pieces, the outer jacket, which carries all axial stresses, and a liner, which constitutes the running barrel. The cylinder heads are of un-

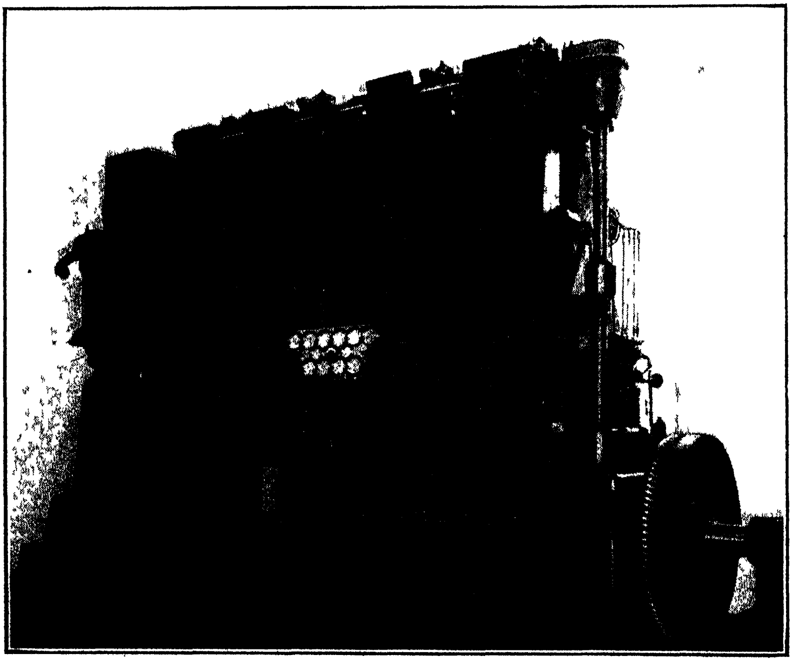


Fig. 181.—1,800 B.H.P. Busch-Sulzer Bros. 2-Cycle Marine Engine

usually simple design and do not contain any scavenging or exhaust valves.

Scavenging air enters and the exhaust gases are discharged through the ports in the cylinder wall, which ports are opened and closed to the cylinder by the piston uncovering the ports on the down-stroke and covering them on the up-stroke, near the end of its stroke.

On the scavenging side of the cylinder there are two tiers of ports. The upper tier is controlled by a timed rotary

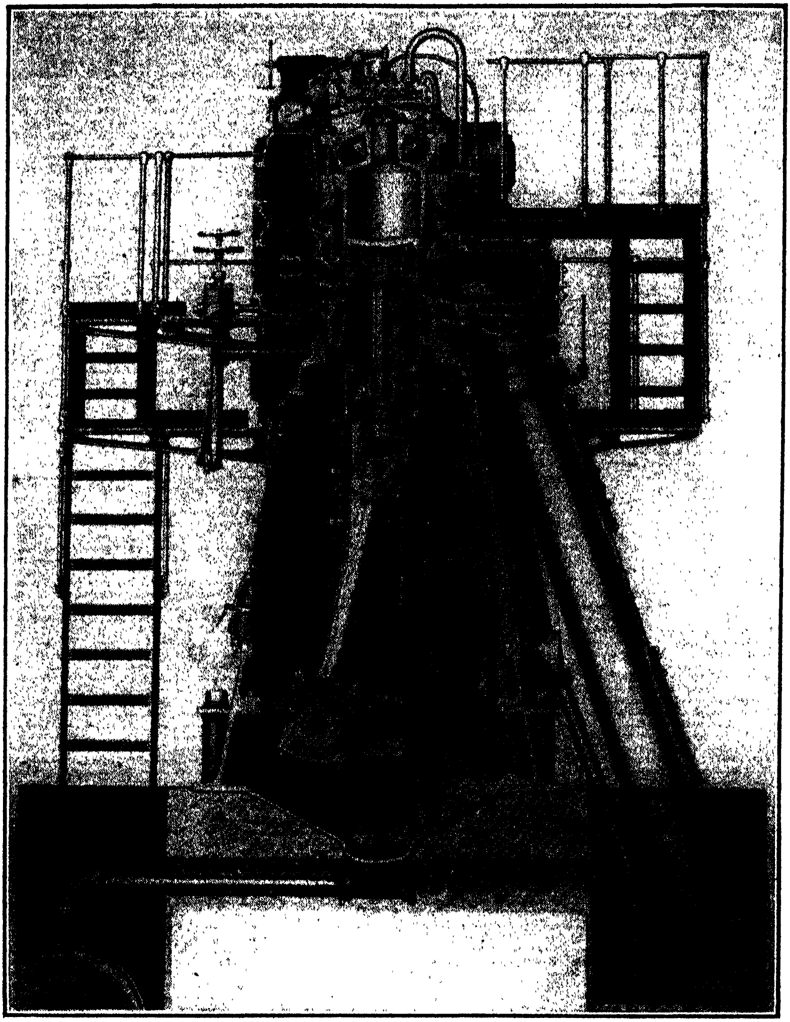


Fig. 182.—Sectional View of Busch-Sulzer 2-Cycle Stationary Engine

valve, driven from the vertical cam shaft of the engine. The lower tier has a free opening into the scavenging air receiver.

The cylinder heads are of special medium hard cast iron. They are of simple, symmetrical design, the head contain-

ing only one central opening of relative small diameter, to receive the combined fuel valve and starting valve cage, thus insuring freedom from casting and heat stresses. The heads do not contain any scavenging or exhaust valves. The cylinder head is rigidly bolted to the top of the cylinder jacket,

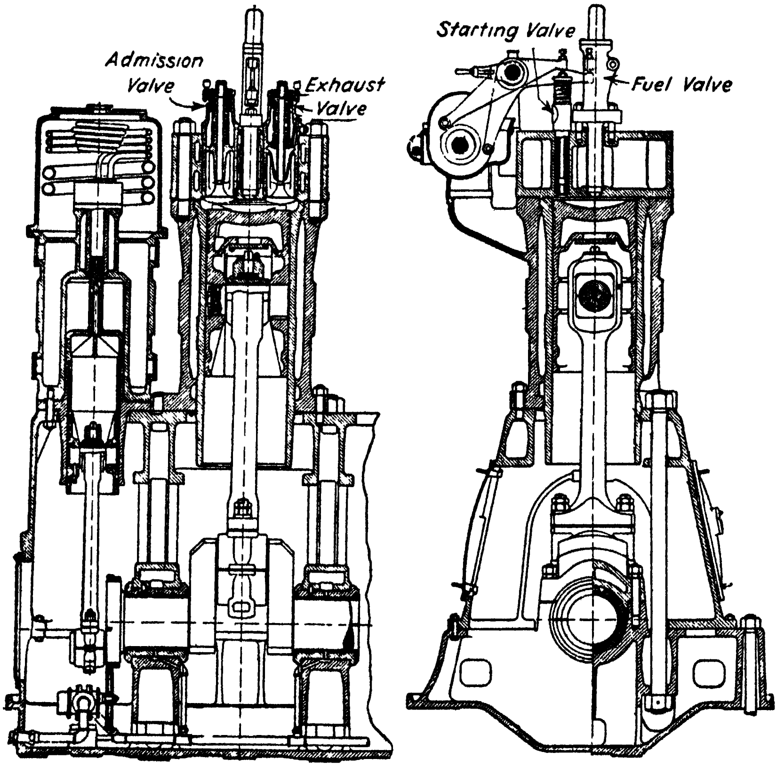


Fig. 183.—Sectional View of Air Compressor and Working Cylinder of Busch-Sulzer Bros. 4-Cycle Stationary Engines

with a registered fit on the cylinder liner. The under side, or combustion space side, of the head is concave, forming, in conjunction with the concave top face of the piston, a symmetrical combustion space of ideal shape.

The camshaft, carrying the cams for operating the valves, extends in front and along the tops of the cylinders, in an en-

tirely enclosed casing, and is driven from the crankshaft, at engine speed, through a pair of helical gears at the lower end of the vertical shaft and a pair of bevel gears at the upper end. This drive is located at the flywheel end of the engine, and taken off the main crankshaft on the flywheel side of the first journal, where it is least subjected to torsional irregularities which might affect the operation of the gears and the governing of the engine.

The marine engines are provided with double sets of starting and fuel cams, and the necessary levers and gear to permit the direction of rotation of their crankshafts to be promptly reversed. The gear is air operated and interlocking devices are provided to safeguard the engine against being started or reversed with the gear in improper position.

The engines are fitted with overspeed governors to prevent racing, by cutting off the supply of fuel to the cylinders when the speed exceeds a predetermined limit, for which the governors may be adjusted.

The fuel pump is of the multiple plunger type, with one plunger for each cylinder, and operated from the vertical shaft.

Additional details on the constructional features of these engines will be found in the Chapters on Engine Details.

Busch-Sulzer 4-Cycle Stationary Engines. Sectional views of these engines are shown in Fig. 183. They are built in various sizes up to 520 B.H.P. and are all of the four cylinder type. The crank cases are of the enclosed box-frame type and are bolted to the bed plate over its entire length.

Standard Fuel Oil Engines. These engines are built in both the vertical and horizontal design and are of the stepped-piston, 2-cycle, port scavenging type. A sectional view through the working and scavenging cylinder of the vertical type engines is shown in Fig. 184.

The interior of the crank case is accessible to an unusual extent as the door openings are both exceptionally high and wide. The greater width is made permissible because of the use of through bolts for carrying the tension strains while the extra height results from the adoption of the stepped piston design. Not only can the main and crank pin bearings be got

at with freedom but the piston pin and its bearing may be removed and adjusted without lifting the cylinder head or removing pistons.

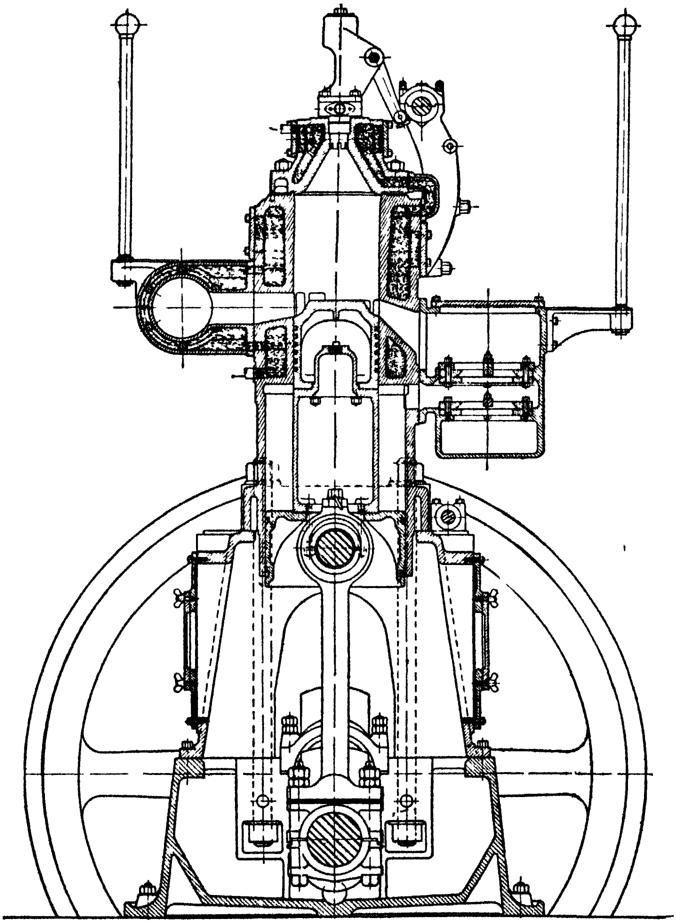


Fig. 184.—Sectional View of Standard Fuel Oil Co.'s 2-Cycle Engine

The removal of the working pistons is facilitated due to the cylinder head being small and comparatively free from valves and piping, and by the method employed for connecting the main piston barrel to the scavenging piston. The piston can be withdrawn by removing the cylinder head and four

easily removable bolts which bolt the working piston to the scavenging piston. By this method the connecting rod and its bearings need not be disturbed. The scavenging piston can be removed from below.

The fuel pump of this engine is of the variable stroke type and is driven direct by a Rites type of inertia governor, the same as commonly employed for operating the valve of a simple steam engine.

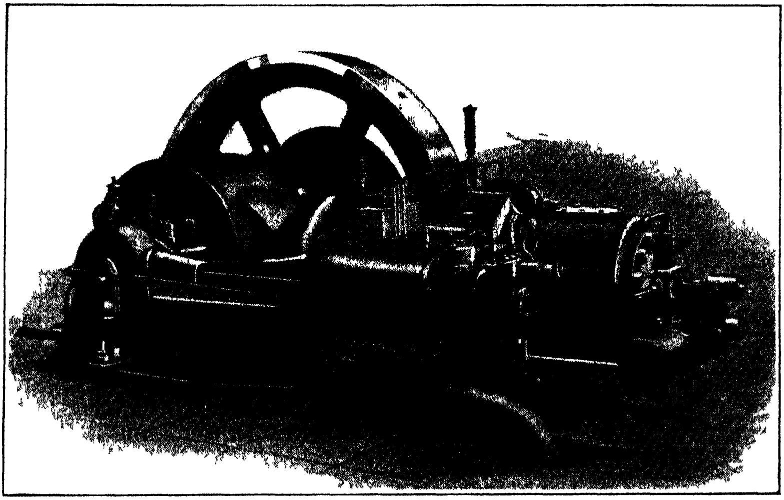


Fig. 185.—Standard Fuel Oil Co.'s Horizontal Engine

The air pressure for scavenging the main working cylinders is generated in the annular space above the crosshead and scavenging piston and around the main piston barrel. The valves for both suction and discharge are of the plate type and operate under very slight differences in pressure. As the areas through the valves and through all ports and passages are very liberal, the scavenging air is handled with the minimum amount of work, and as a result the overall mechanical efficiency of the engine is high, comparing very favorably with the best 4-cycle practice. The scavenging manifold which extends across the front of the engine contains the pockets in which are located the suction and discharge valves of the

scavenging pumps, also the passage through which the air is distributed to the working cylinders.

The vertical engines are made in sizes of from 100 to 900 brake horse power, and with from 2 to 6 cylinders. The horizontal engines are of the one and two cylinder type. A single

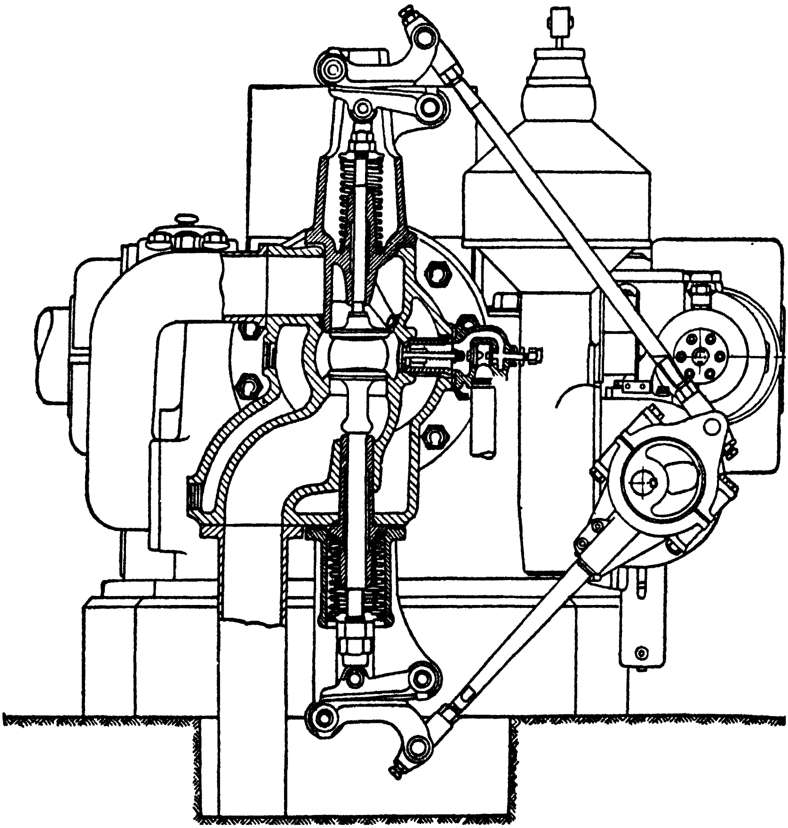


Fig. 186.—End Section of Allis-Chalmers Engine

cylinder 75 horse power, horizontal engine is shown in Fig. 185.

Allis-Chalmers Engines. This company builds a horizontal engine having eccentric driven admission and exhaust valves. A horizontal compressor is mounted on the side of the engine frame. They use the Lietzenmayer system of open fuel noz-

zle, and variable stroke plunger fuel pump. Fig. 186 shows an end section and Fig. 187 a longitudinal section of these engines.

Lombard Diesel Engines. These engines are of the medium speed, heavy duty, 4-cycle type and are of late design. A sectional view of the four cylinder engine is shown in Fig. 188.

Each cylinder with its cylinder head is cast integral. This construction eliminates the cylinder-and-head joint, and the mass of metal required for it, from the heat and pressure

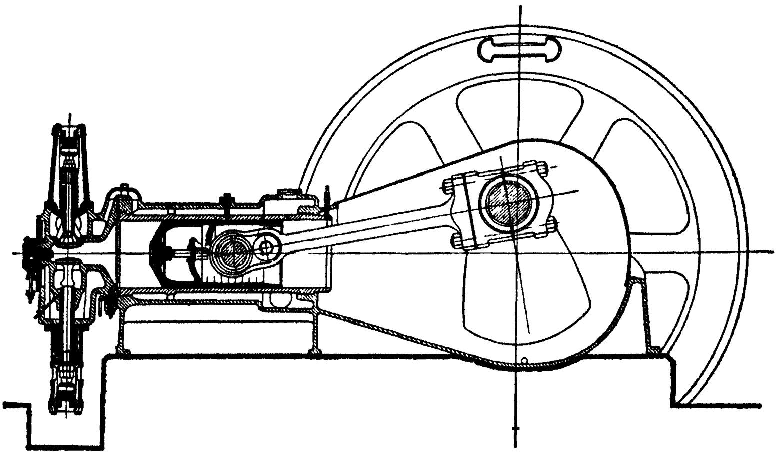


Fig. 187.—Longitudinal Section of Allis-Chalmers Engine

changes of the combustion chamber and thus provides ample and unobstructed space for the circulation of cooling water, and secures the uniform expansion and contraction necessary for long life.

A detachable skirt bolted to the bottom of each cylinder serves as a guide for the piston and the metal-to-metal joint between the cylinder and skirt is subjected only to exhaust temperature and pressure. With the crank on bottom center and the skirt detached from the cylinder, any piston with its connecting rod can be swung forward through the crank case door opening, without removing the cylinder head, disconnecting the crank pin bearing, dismounting the valve gear, or otherwise disturbing adjustments.

Each piston is fitted with seven compression rings and one wiper ring. When the piston is on bottom stroke the skirt can be dropped down and instant access had to the piston rings.

The bed plate is heavily ribbed to rigidly support the crankshaft bearings. Upon it are mounted the individual rear frame sections, one for each working cylinder, and the air compressor frame. Front stanchions and tight fitting doors and covers complete the crank case enclosure. Four heavy steel tie rods, which extend from each cylinder to the bed plate, carry the working load and relieve the frame sections of all tensile stresses.

By removing the tie rods and stanchions, and crank case doors along the front of the engine, the crankshaft may be removed without dismounting any of the cylinders or disturbing the valve gear.

The cam shaft, supported in bronze bearings, is enclosed in a continuous housing mounted centrally over the engine cylinders with cams and rocker rollers running in oil. All valves are steel forgings seating in removable cages and are operated by simple rocker levers. Admission and exhaust valves and cages are interchangeable and the exhaust valve cages are water cooled. The arrangement of cams and the shifting of the cam shaft provide for the quick starting of the engine with air from the air storage tanks and, as the engine attains speed, for the change from air to fuel in the several cylinders successively. With the marine engine, which is very similar to the stationary type, the cams are arranged for ahead and astern maneuvering through the shifting of the cam shaft.

The air compressor is of the balanced duplex design; with plain, unjacketed cylinders mounted in an ample water box, which also contains the inter-cooled coils. All valves open outward, and are easily accessible.

Nordberg Diesel Engines. Fig. 189 shows a cross sectional view of the Nordberg 2-cycle, overhead valve scavenging, cross head type Diesel engine. These engines are built in sizes of from 200 to 4,000 B.H.P. The range being covered by three types of engines, the general arrangement of each being about the same.

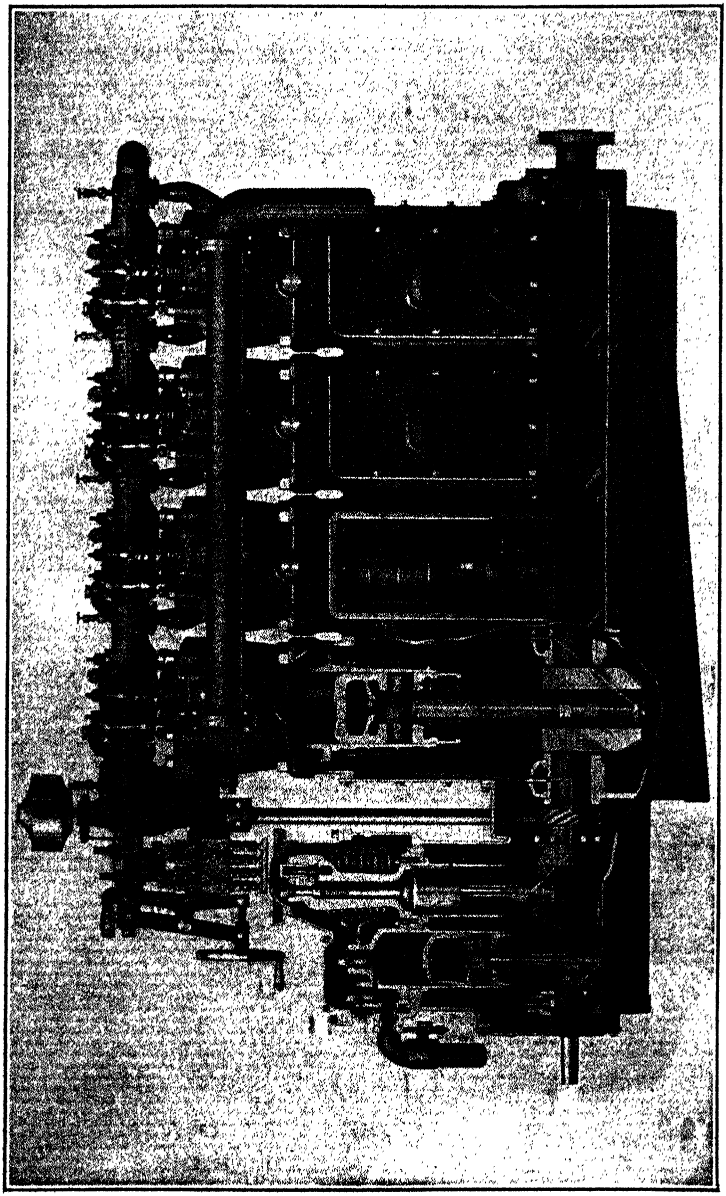


Fig. 188.—Sectional View of Lombard 4-Cycle Engine

The spray air compressors are of the three-stage type, having specially designed plate valves and pistons built in steps. Two types of compressors are used. On the smaller engines the compressor is driven from the scavenging pump by means of a beam coupled to the scavenger pump cross-head. The larger engines have the compressor directly driven from the crankshaft.

The working pistons are water cooled; the water being conducted to the pistons by means of a swinging joint. It is made in two sections; the upper section contains the piston rings, and the lower section form a sleeve which keeps the exhaust ports covered, until the time for exhausting the gases.

The cylinder heads are made of special cast iron and are fitted with four scavenger valves, one spray valve, and an air starting valve. This concern is one of the first to successfully design an overhead valve scavenging 2-cycle engine and overcome the difficulties of cracked cylinder heads due to the great temperature range to which the head is subjected to. The valve gear need not be disturbed when detaching a cylinder head as all parts lift away from the cams.

The bearings supporting the cam shaft are ring oiled and the eccentrics on the cam shaft are supplied with large oil wells, both of which require a minimum amount of attention. The cams are rigidly secured to the shaft by a specially designed Nordberg method which eliminates the use of keys and, when necessity requires, are easily detached. In case of removal, the novel construction prevents replacement of any cam except in exactly the same position as it was prior to its removal.

The crosshead is a forged piece to which the piston rod is bolted, and has two pins to which the forked end of the connecting rod is attached. The water for cooling the piston enters through one duct in the piston rod and down through another, and thence out the other crosshead pin. The crosshead shoes are cast steel members bolted to the crosshead proper and have shims under them for adjusting the clearance.

The connecting rod is a solid forged piece to which a cast steel crank pin box is bolted. The portion which is

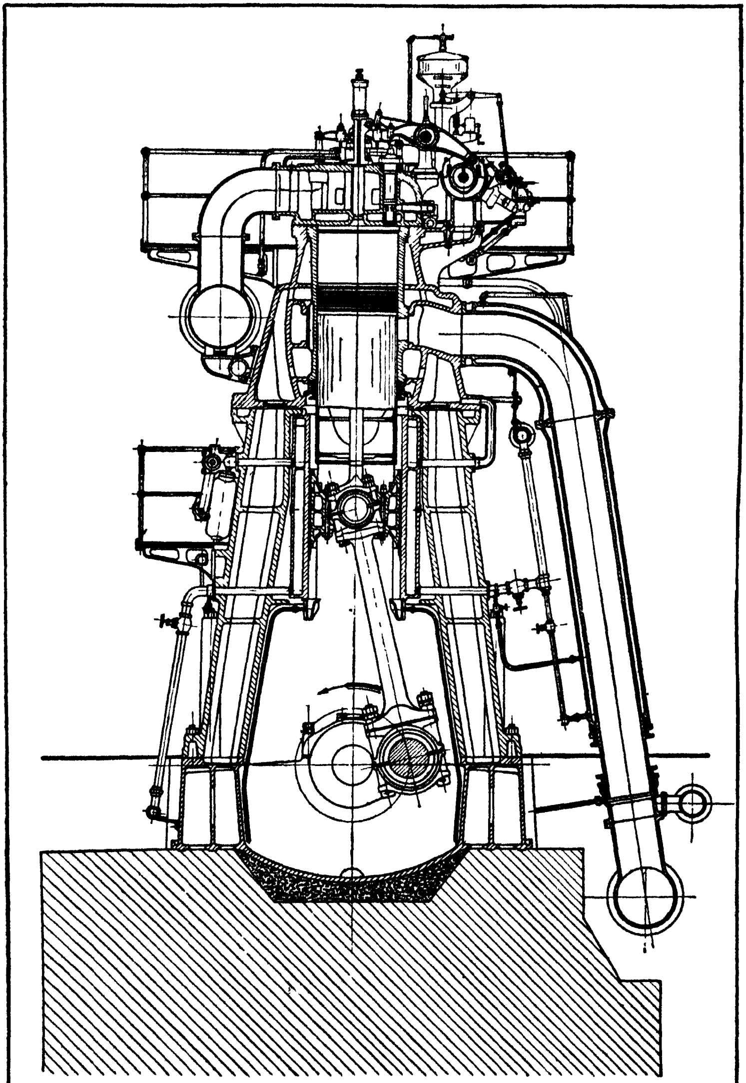


Fig. 189.—Cross-Section of Nordberg, Overhead Valve Scavenging 2-Cycle, Stationary Engine

attached to the crosshead is forked and has two cast steel boxes. Both the crosshead and crosshead boxes are lined with an especially hard babbitt.

The adjustment of both the crankpin and crosshead pin boxes is made by means of shims. The pressure on the connecting rod is limited to the compression only, inasmuch as the engine works on the two-cycle principle and is single acting. This eliminates all work down by the crankpin and crosshead pin bearing bolts. Each crosshead pin bearing and the connecting rod is supplied with special lubricating pump which is driven by means of a link.

The scavenging and air starting valves are made of alloy steel and the seat of the scavenging valves consists of a specially constructed loose member which can be replaced when necessary. The scavenging valves seldom require attention and never become heated as cold scavenging air passes over the valves once each revolution. The valve seat and cage are removed in a single unit.

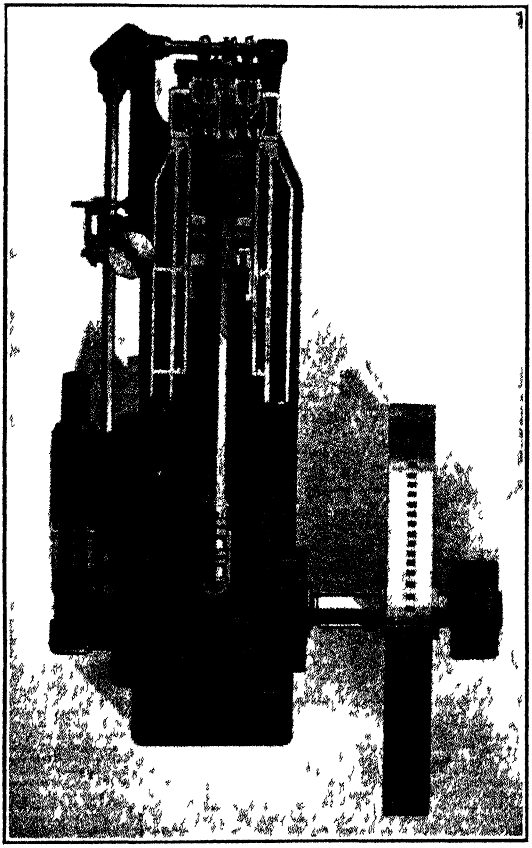
The air starting valve is made of non-corrosive material, the cage and valve being built in a single unit. The construction is such that even when the valve remains idle for a long period of time it will not stick.

One fuel pump is used for each cylinder and they are built in units of two to make the drive as simple as possible. A means is provided so that any fuel pump can be cut out independently of any other pump. It is equipped with the usual discharge and suction valves. The valve which acts as a by-pass valve is under the control of the governor and reduces to a minimum the reaction on the governor. The by-pass valve is so located as to relieve the fuel pump of all air. Each pump is equipped with a hand pump for priming the fuel lines. Means are also provided so that both suction and discharge valves can be lifted simultaneously and the fuel pump flooded.

The Nordberg Company has succeeded in developing a very successful fuel atomizing valve, capable of properly atomizing fuel varying from very light fuel oil to a high asphaltum fuel such as Mexican fuel oil down to 12 degrees Baumé. All parts of the fuel atomizing valve are very easily

inspected and the needle can be removed without dismantling any part of the valve proper. The use of intricate passages has been reduced to a successful minimum.

The frame with cylinder for the smaller types is cast in one piece. There are no complicated cores and the metal is



Sec. 190.—Sectional View of De La Vergne Horizontal Engine

uniformly distributed throughout the casting. The cylinder barrel is cast separately in the form of a liner which is pressed into the cylinder casting and is removable. Numerous handhole covers on the cylinder are provided for cleaning out the water jackets. The crosshead guides and part

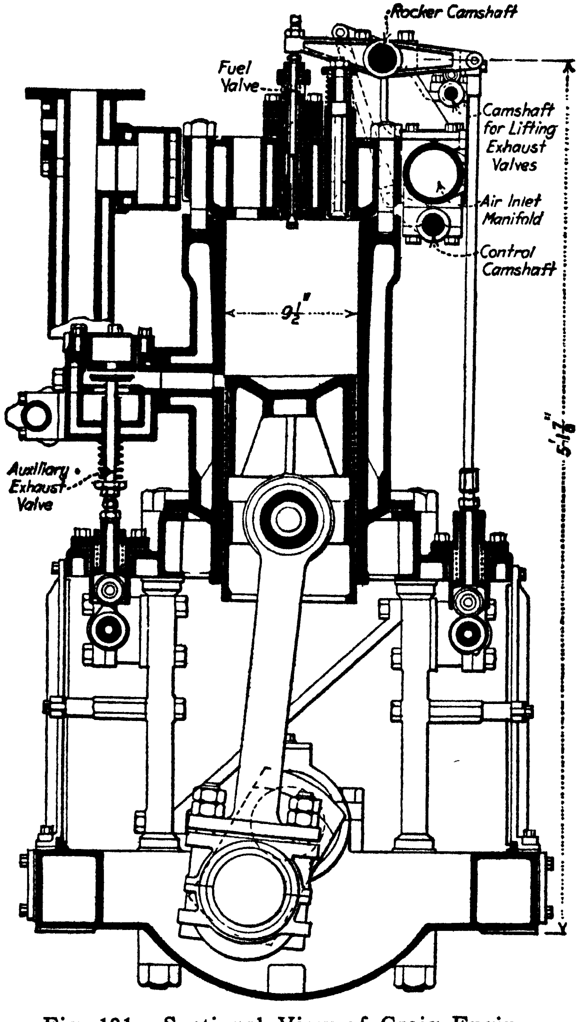


Fig. 191.—Sectional View of Craig Engine

of the cylinder into which the liner is pressed are bored at the same time, hence, perfect alignment is secured. The cylinder head forms a joint on the liner and is held in place by means of special alloy steel studs.

De La Vergne Diesel Engines. These engines are of the horizontal, single acting, 4-cycle type, built in units of from

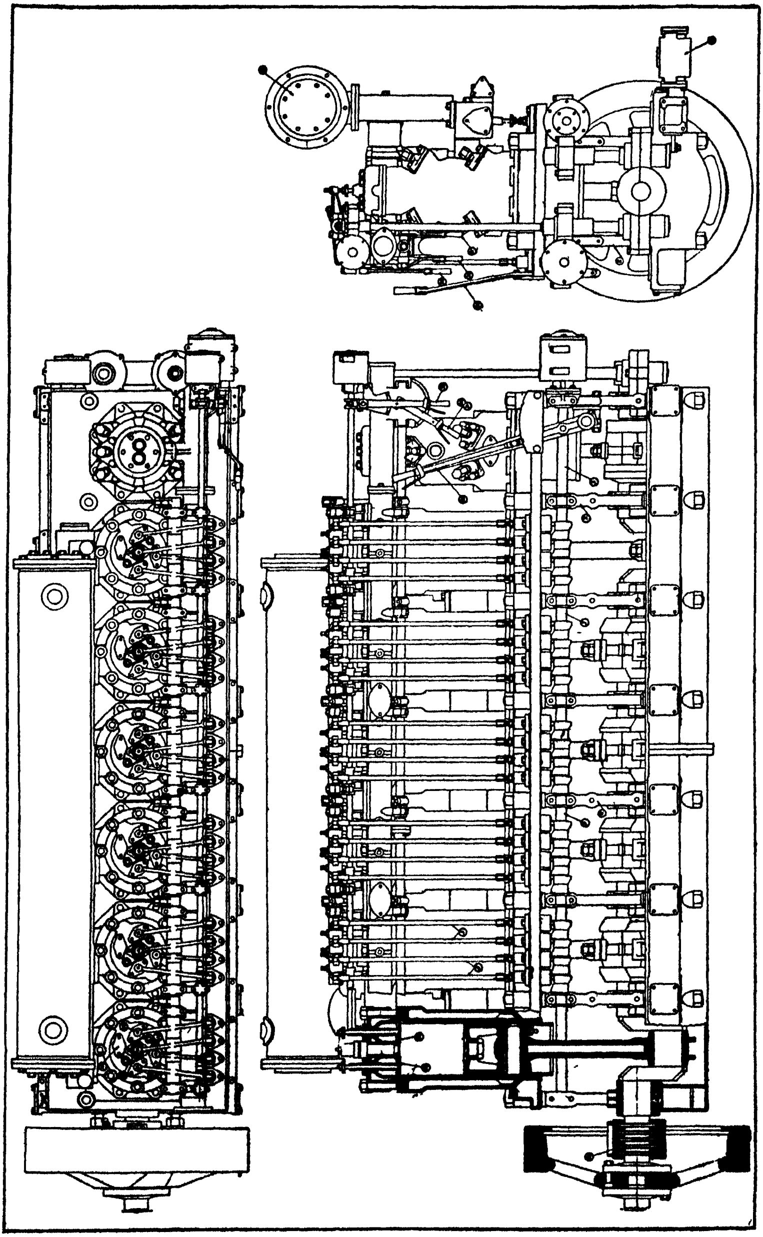


Fig. 192.—Plan and Elevation of Craig Engine

one to six cylinders. A horizontal cross section of the 180 B.H.P. single cylinder engine is shown in Fig. 190. The cylinder is fitted with a removable liner inserted into the frame from the head end and held in position by pressure between the cylinder head and frame. The crank case end of the liner passes through a simple rubber packed joint and is free to change its length with changes in temperature.

Spray air is furnished by a double acting 3-stage compressor, directly driven from the crankshaft. The inlet and exhaust valves are mounted in removable cages, both valves and cages being removable. Removable cast iron seats are used for these valves.

Mounted across the back of the cylinder head are brackets carrying a shaft on which pivot the rocker arms operating the valves. The rocker arm opening the spray and starting valves pivots on an eccentric sleeve also rotating on this shaft.

Valve cams are mounted on the shaft running across the head of the engine. These are driven by bevel gears from the layshaft which in turn is driven from a skew gear on the crankshaft. On the layshaft are mounted cams for operating the fuel pump and skew gear for driving the governor. All camshaft and layshaft bearings consist of bronze boxes, chain oiled and all gears are completely enclosed and run in oil.

The Craig Marine Engine. A sectional view of the Craig Diesel engine is shown in Fig. 191 which shows the distinctive lower exhaust port system which is a distinguishing feature of the Craig engine. A plan and elevation view is shown in Fig. 192.

With a view to providing accessibility and lightness together with rigidity, the frame is made up of stanchions with cross-bracing, mounted on a one piece bed plate of box section and supporting a cylinder table upon which are carried the working cylinders and the compressor. Removable splash guards are fitted at the front and back.

Heat conditions have been minimized by providing auxiliary exhaust ports at the lower ends of the cylinders, thus releasing the exhaust gases near the end of the power stroke. The exhaust valves in the cylinder heads are thereby relieved

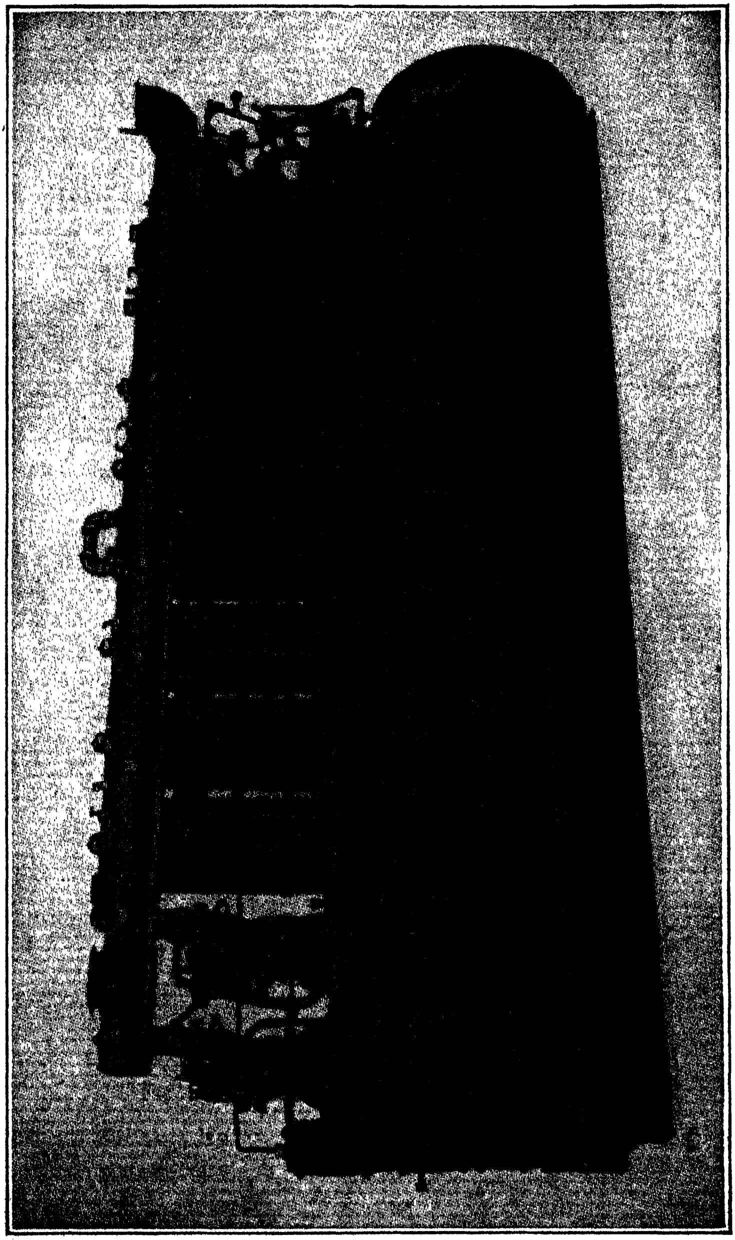


Fig. 193.—240 B.H.P. Nelseco Engine

of much of the heat to which they would ordinarily be subjected; they are in contact with the water cooled seats for a longer period and do not open against internal pressure. Also the piston is exposed to the intense heat for a shorter time. By thus diverting the bulk of the exhaust gases from the exhaust valves, it has been possible, in the engines above the 200 horse power rating, to dispense with the usual detachable cages, thereby leaving room to provide for two exhaust valves of ample area.

Each of the lower exhaust passages has an exhaust valve operated by a cam shaft at the back of the engine. This valve is closed during the suction stroke and compression stroke, but is wide open by the time the piston uncovers the exhaust port on the working stroke. In this way exhaust gases are prevented from being sucked back into the cylinder on the suction stroke and the valve is relieved from opening under pressure.

Each cylinder has a fuel injection valve, two air inlet and two upper exhaust valves, each pair being operated by a single rocker; an air starting valve, and a relief valve. With the exception of the last named valve they are all of nickel steel. The rocker shaft for each cylinder is separate and carries two eccentrics, one for the air starting valve rocker and the other for the fuel injector valve rocker. The camshaft brackets and valve operating gear are carried on the air intake manifold.

Salt water is drawn through the engine bedplate by a bronze pump of the piston type carried at the end of the bedplate, and is supplied first to the compressor, thence to the lower exhaust manifold. With the ample cooling spaces provided, it is possible to lay the hand on any part of the engine without discomfort. The pistons are not cooled, but the main bearings are.

A very simple system of lubrication is provided, which also makes for cleanliness. The main crankshaft and lower connecting road bearings are supplied by wick feed and a wiper on the crank face. Sight feed lubricators, similar to those employed on many steam engines, feed oil to each cylinder at a single point. This oil is picked up by a special

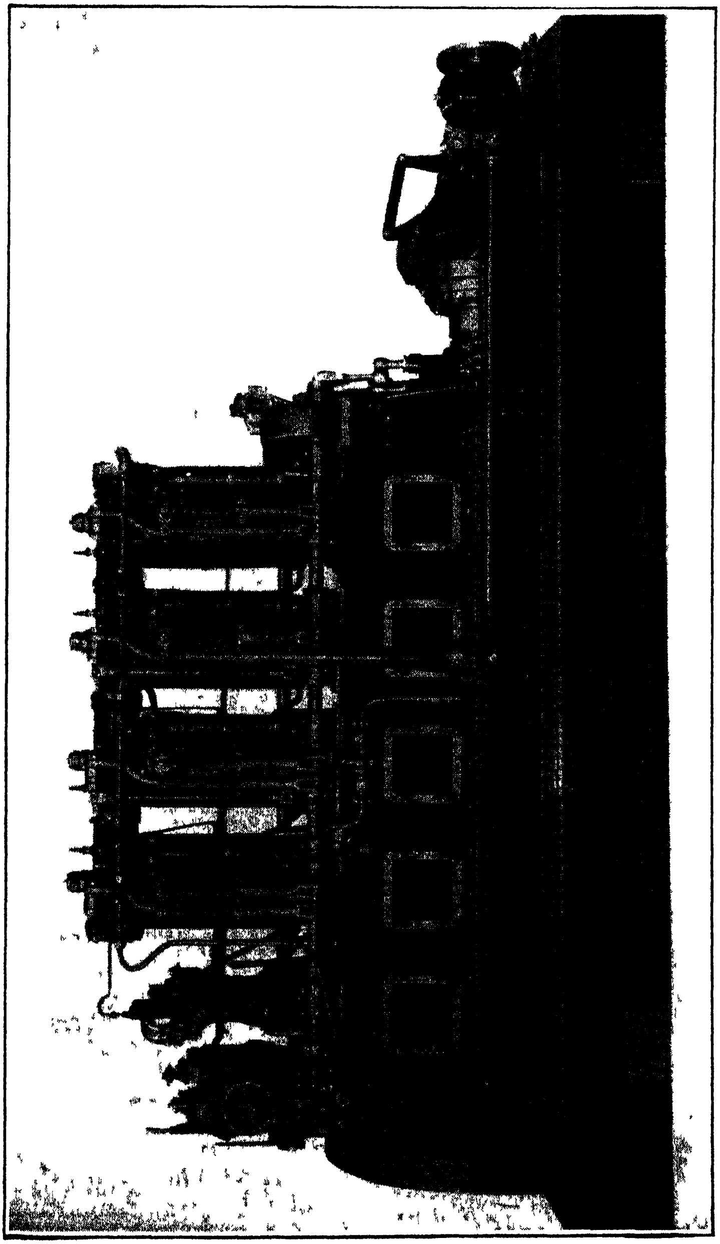


Fig. 194.—120 B.H.P. Nelseco Engine

piston ring, which distributes it over the inner surfaces of the cylinder and also through holes in the piston to the wrist pin bearing, the wrist pin being fixed in the piston with the bearings at the connecting rod end. The camshaft bearings, rocker arms, push rods, etc. all have oil holes which are kept supplied by the operator.

The Nelseco Engine, shown in Fig. 193 and having eight working cylinders, two 2-stage air compressors and working on the 4-cycle principle, is a type that has given excellent service aboard American submarines. They are made with four, six and eight cylinders and develop 120, 180, and 240 brake horse power respectively. The 120 horse power engine is shown in Fig. 194 and a cross section of one of the working cylinders in Fig. 195.

2,500 H. P. McIntosh and Seymour Engine. The McIntosh and Seymour Corporation recently completed a large marine engine of 2,500 Indicated Horsepower for marine work, which is illustrated in Fig. 196.

This engine is of the vertical, single acting, four-cycle, crosshead type and is directly reversible. There are six working cylinders, each of thirty inches bore and forty-eight inches stroke, and at a speed of 115 R.P.M. gives a rated shaft horse power of 1915.

The compressor is of the three stage type and is mounted on the forward end of the engine, while at the aft end is mounted the thrust bearing which is of the Kingsbury type.

The engine is built to comply with the rules and regulations laid down by the various Classification Societies operating in this country; principal of which are the American Bureau of Shipping and Lloyds' Register of Shipping. The scantlings of the various parts, therefore, are such that they come within the rules of either society, so that it is only a matter of arranging for physical tests, for these engines to be built directly under either societies' rules.

The weight of the engine is approximately 318 tons which gives a weight per indicated horsepower of 315 pounds.

The engine is so designed that right and left hand engines can be easily built and so, twin screw arrangement in the ship can easily be taken care of.

The maneuvering gear is at the forward end of the engine and is electrically operated.

The general arrangement of the engine follows closely that of the smaller types of marine engines which have been

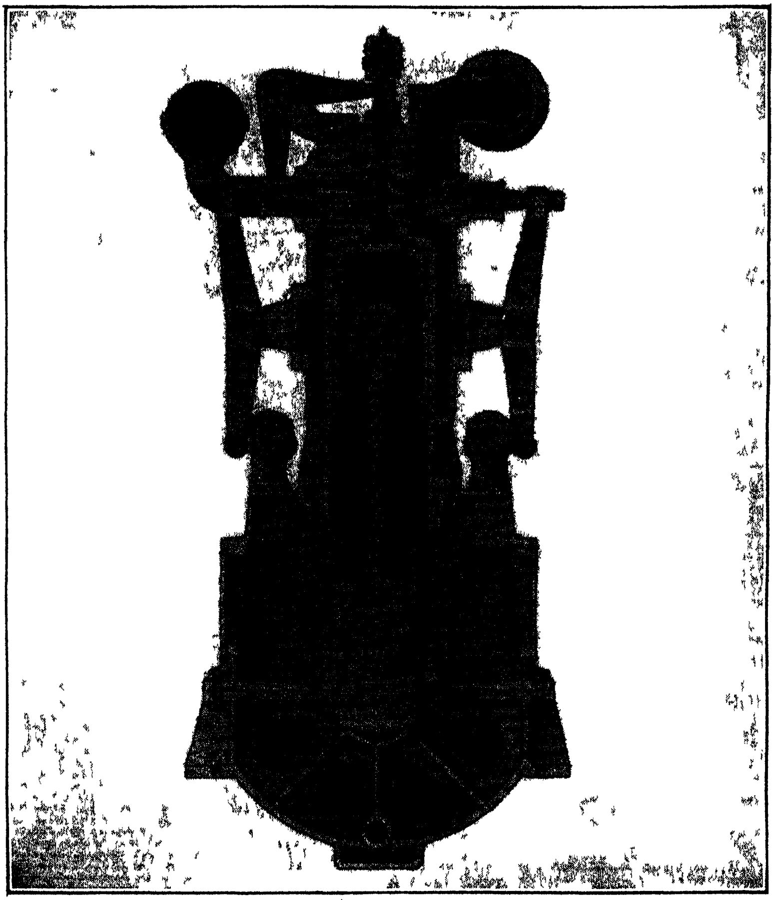


Fig. 195.—Cross-section of Nelseco 30 B.H.P. Cylinder Units

built by this Corporation for so many years and which are in successful operation in a large number of American ships.

The base of the engine is made in two parts and bolted together at approximately the center. The forward half is

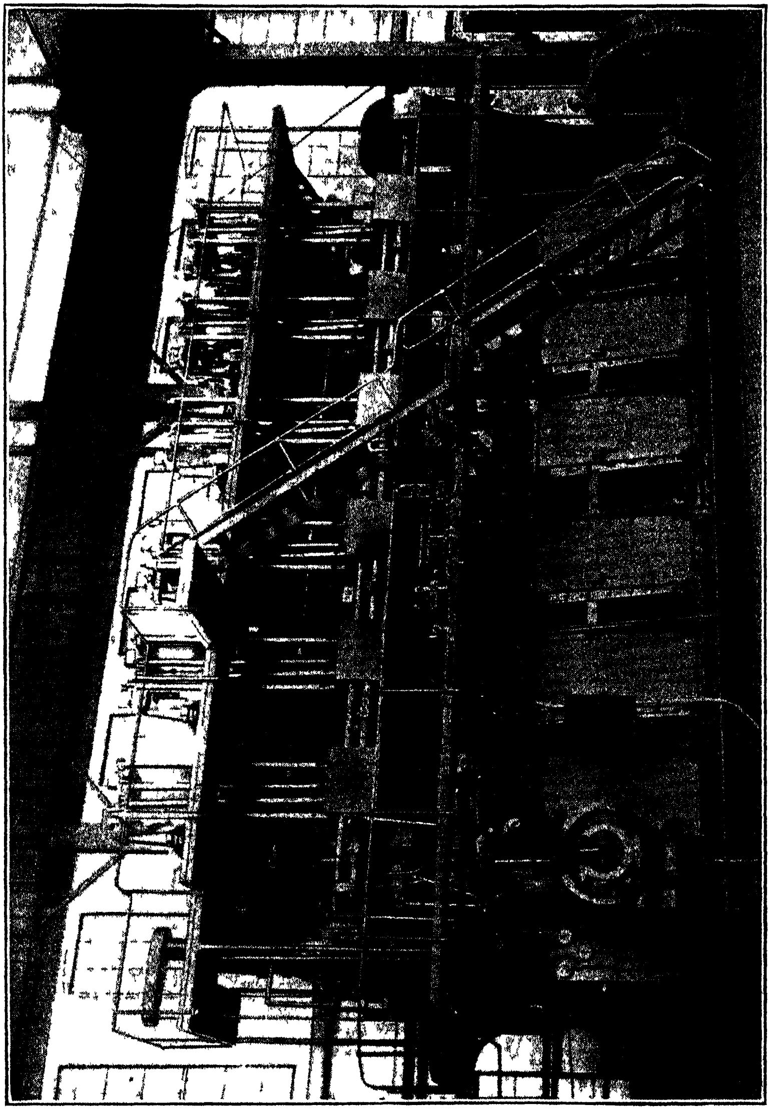


Fig. 196.—2,500 H.P. McIntosh and Seymour Marine Engine

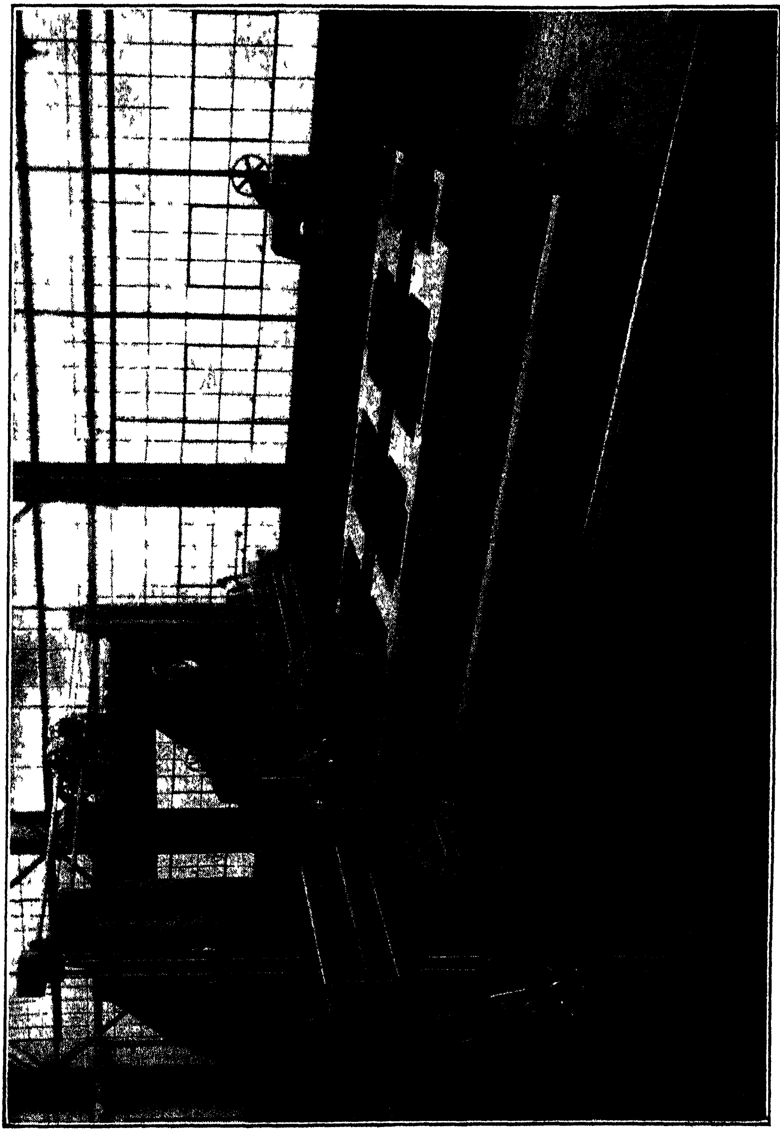


Fig. 197.—After Half of Base on Planer, McIntosh and Seymour 2,500 H.P. Engine

arranged to carry four main bearings and the compressor, while the aft half takes the other three main bearings and the Kingsbury thrust bearing. Fig. 197 shows this latter piece on the planer in process of machining.

Special arrangements are provided to keep these two halves exactly in line so there can be no possible trouble arising from this cause due to any possible "working" when in the ship.

The bolting for attaching the frames is kept as close as possible to the bearings, but at the same time leaving convenient access to the main bearing caps. The arrangement, which is carried out throughout the whole of the McIntosh & Seymour designs, makes them have a minimum length of bearing girder, and insures great stiffness as well as strength with minimum weight.

The ends of the bearing girders are carried by longitudinal members and these bearing girders are cast integrally with the enclosed base, the solid bottom and longitudinal members all making a base of great stiffness and longitudinal strength.

An idea of the sizes and construction of the frames can be gathered from Fig. 198, which shows a set of these frames in course of manufacture while Fig. 199 shows a single unit.

The substantial design of these frames is self evident, but at the same time is such that weight is reduced to a minimum without interfering with the strength of the structure. They are arranged for the mounting of the cylinders on their upper ends and to carry the principal stresses from the cylinders to the base in the most direct manner possible, and at the same time are amply stiff enough to take care of all transverse stresses throughout the machine and they are connected on one side by the crosshead guides and on the other by tie plates so as to give the necessary longitudinal stiffness to the frame work, and at the same time to give convenient access to the working parts.

Sheet metal covers are provided over the crank openings and fitted with sliding doors to give convenient access for inspection while the engine is in operation without any chance for throwing lubricating oil on the gratings.

The cylinder head is made of a deep cylindrical section,

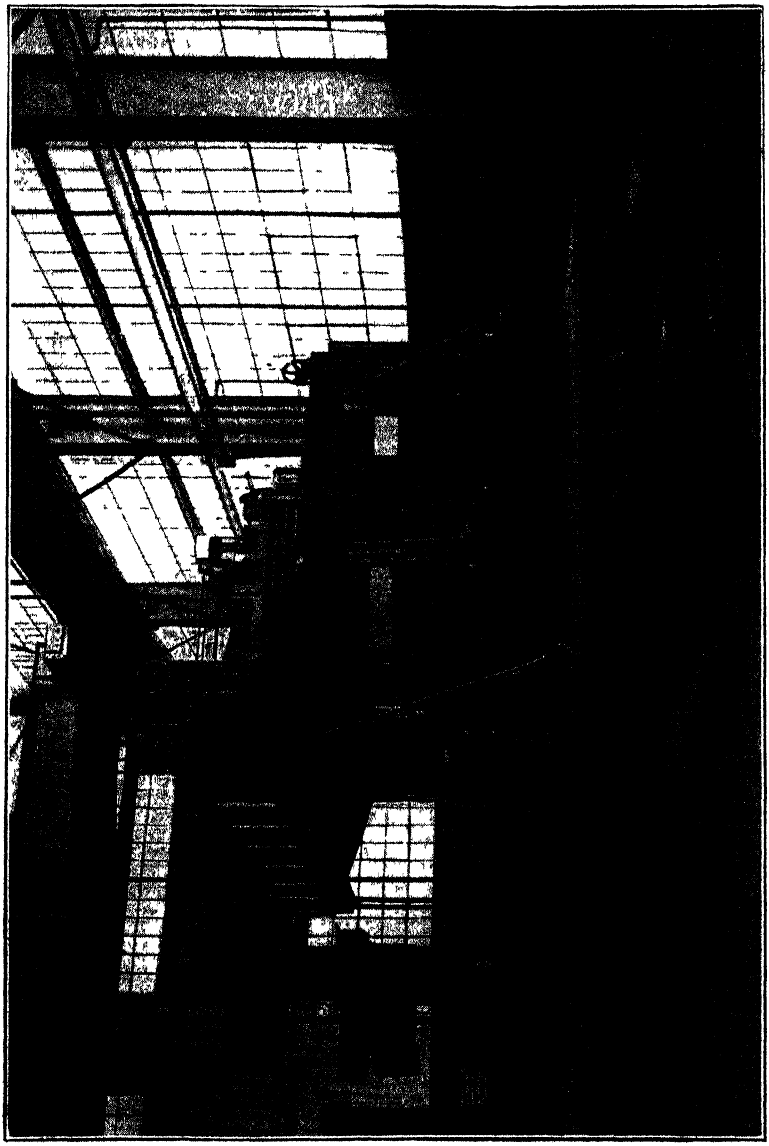


Fig. 198.—One Set of Frames on Planer, McIntosh and Seymour 2,500 H.P. Marine Engine

cast in semi-steel giving strength and stiffness with the minimum metal thickness, and is of the same type as that shown in Fig. 172.

The bottom member is only the diameter of the liner fit leaving large openings between the studs, corresponding with the openings in the cylinder jacket, thus providing for ample

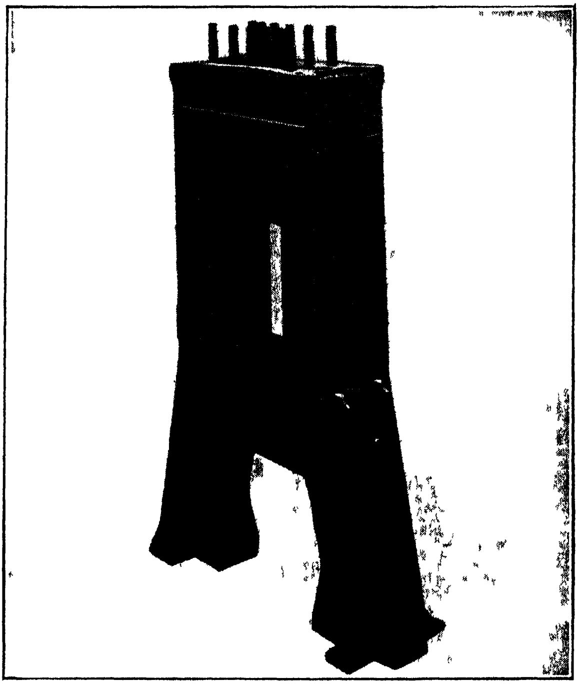


Fig. 199.—Frame Between Cylinders, McIntosh and Seymour 2,500 H.P. Marine Engine

jacket spaces and the fullest possible flow of the cooling water, which in connection with the moderate heat conditions due to the complete combustion of the fuel, and the special material mentioned have proved to give this head extremely long life in service.

Each head is fitted with suction, exhaust, starting air and fuel valves. The suction and exhaust valves are interchangeable the heads being of cast iron fused to a steel stem. These

are fitted into renewable cages (Fig. 200) which are also interchangeable and with springs of such length as to give the maximum life. The valve tappet connects to the upper end of the valve stem.

The fuel valve which is the most important part of a Diesel engine, being fitted with the Hesselman type pulverizer (Fig. 176) which is claimed to be the most effective means

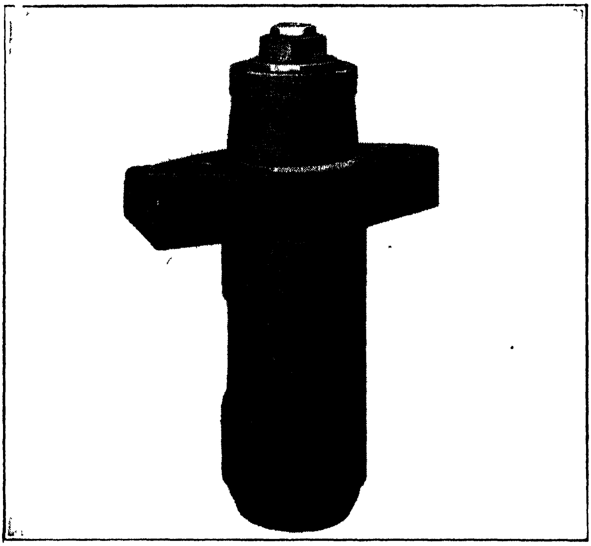


Fig. 200.—Air Inlet and Exhaust Valves, McIntosh and Seymour 2,500 H.P. Marine Engine

so far devised for the preparation of the fuel for burning in a Diesel engine cylinder. This pulverizer gives maximum efficiency with prompt and complete combustion and does not need to be varied with the different grades of fuel that are met with from time to time.

Where separate pulverizer cones and plates are used in connection with fuel spray valves the grooves in the cones and the holes in the plates or the number of plates fitted have to be varied to meet the varying conditions of the fuels from time to time.

With the Hesselman pulverizer, these varying conditions are of non-effect, the same pulverizer being suitable for the heavy Texas "C" boiler oil as for the lighter oils of approximately 30 to 32 degrees Baumé gravity.

The cylinders which are clearly shown in photograph Fig. 201 carry the main stresses through a substantial jacket, on the lower side of which is cast a heavy flange for connecting

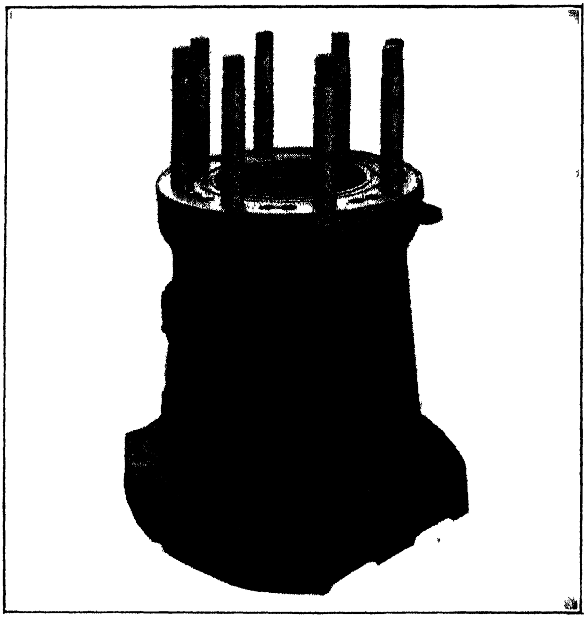


Fig. 201.—Cylinder of 2,500 H.P. McIntosh and Seymour Marine Engine

the cylinders to the frames. This jacket is fitted with a renewable liner made of a special mixture of charcoal iron, which the Corporation claims gives the best wearing qualities.

This liner is pressed into the jacket at its upper end while the lower end is free to expand and contract as necessary.

Ample openings between the cylinder studs are arranged to allow for the free circulation of the cooling water at the hottest part of the cylinder and hand holes are provided on

the outside of the jacket to give convenient access to the jacket space for cleaning purposes.

This engine being of the crosshead type, the pistons are no longer than is necessary to carry the piston rings and are

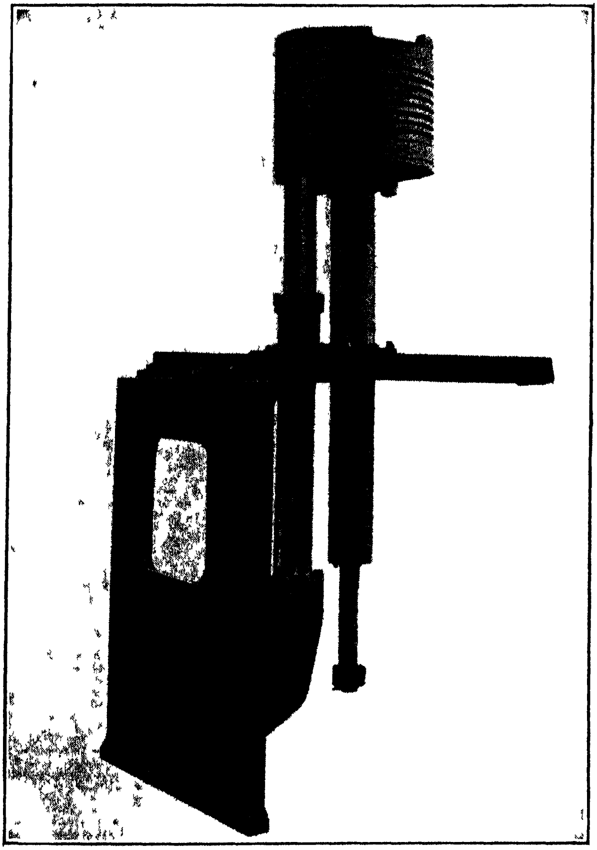


Fig. 202.—Piston and Piston Cooling Assembly, McIntosh and Seymour 2,500 H.P. Marine Engine

securely bolted to a flange at the top end of the piston rod and arranged for water cooling.

They are cast in dry sand and are of a special charcoal iron mixture very similar to the mixture used in the cylinder liners. Accurate machining is a special feature in connection

with their manufacture, and the number of rings fitted are only such as is necessary to give the satisfactory service demanded of a marine piston.

Illustration Fig. 202 shows one of these pistons mounted complete with the piston cooling arrangement on one of the tie plates which are fitted between the "A" frames. As will be seen one set of cooling pipes is bolted securely to the under

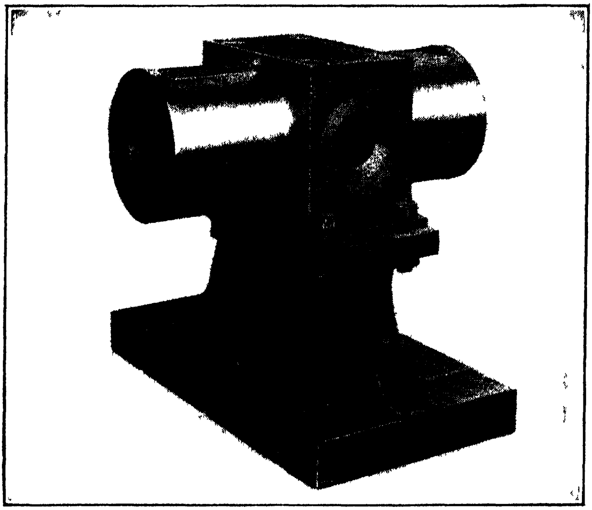


Fig. 203.—Crosshead Slipper of McIntosh and Seymour 2,500 H.P. Marine Engine

side of the piston, and the other set is bolted securely to the tie plate, and the arrangement is such that there are no joints on the inside that would allow the cooling medium to mix with the ordinary lubricating oil.

The telescopic joints in this arrangement are not supposed to be absolutely tight, and any leakage taking place near the piston is provided for and caught by the diaphragm shown, which is provided with a stuffing box for the piston rod.

Such leakage as may take place at the lower end runs into the recess in the tie plate and passes with the overflow water into the bilges. Every precaution is taken to avoid any leakage taking place in these telescopic pipes, and instead

of the usual pipe arrangement being used a special system of triple pipes has been designed for this purpose, which in service has proved to be extremely efficient and well worth the extra trouble of manufacture and fitting.

The piston rod is of open hearth forged steel, the upper end of which is a flange to take the piston, and the lower end is turned down to form a large shoulder where the rod passes through the crosshead, and the end of the reduced portion is

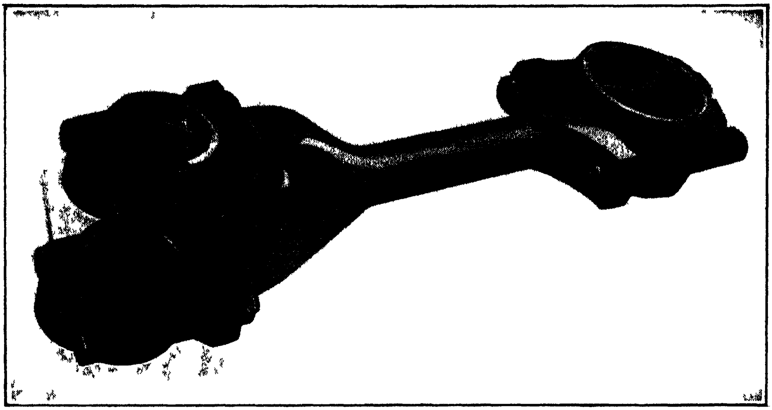


Fig. 204.—Connecting Rod of McIntosh and Seymour 2,500 H. P. Marine Engine

fitted with a nut which is specially secured to avoid any chance of this nut becoming loosened.

A crosshead which is illustrated in Fig. 203 is of the ordinary standard marine type with the pins forged in one with the body. This crosshead is securely fastened to the slipper which is made from a steel casting, the face of which is babbitted. An adjustment for wear of this babbitted face is provided for by liners between the slipper and the crosshead.

The guides for the slipper are made in cast iron and carefully machined, and are arranged for fitting between the frames on the opposite side to the tie plates carrying the piston cooling arrangement. These guides are of the usual marine design and are carefully machined to receive the babbitted slipper already mentioned.

The connecting rod, which is illustrated in Fig. 204 has a body of forged steel and bearings boxes of cast steel lined with white metal. As will be seen this is of the usual marine type construction and of substantial design.

The crankshaft is made in two sections, each section being built up from separate web and crank pin forgings. The two halves are clearly shown in Fig. 205, ready for bolting together at the center.

At the forward end is fitted an overhung crank for driving the compressor, and at the aft end the cam shaft driving gear is mounted.

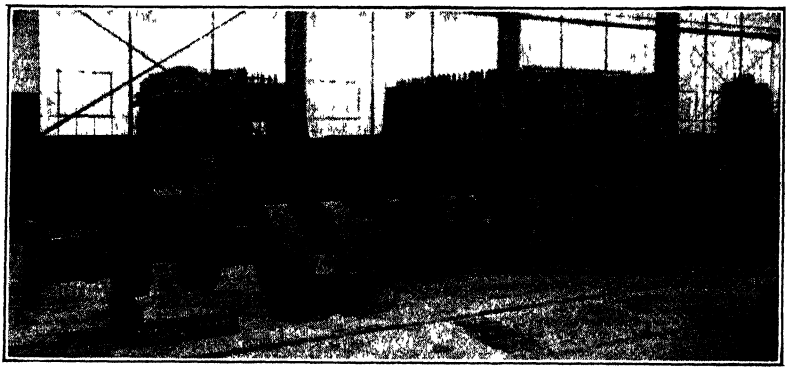


Fig. 205.—Crankshaft of 2,500 H.P. McIntosh and Seymour Marine Engine

The cam shaft is driven by means of spur gearing, these being driven through a train of five wheels, three being idlers. These wheels are made in cast iron and the teeth are cut to special shape so as to give the maximum strength at the root and combined with the width of the wheel gives a long life to this gearing.

The cam shaft runs throughout the whole length of the engine at about the center and has mounted thereon a double set of cams, the second set being used when the engine is reversed and to bring these into operation the cam shaft is moved sideways so as to bring this second set of cams under the valve rods.

The movement of the cam shaft is made through levers operated from the electrical control and maneuvering gear.

The valve rockers are of cast steel and are mounted upon a fulcrum shaft which in turn is seated on brackets fastened to the cylinder head. These rockers are provided with the usual hardened adjusting screw for the necessary valve settings and adjustment.

The fuel pump follows the usual system adopted by McIntosh & Seymour of having a separate plunger for each cylinder. Each one of these separate pumps being driven by an eccentric which is mounted on an inclined eccentric on the pump driving shaft.

This inclined eccentric is moved endwise by a collar arranged so that when the pump shaft is at one extreme all of the pump eccentrics run centrally with the shaft and give no stroke to the plungers. At the other extreme the eccentrics have maximum eccentricity and the plungers have a full stroke corresponding to the maximum delivery of fuel desired.

In addition to the hand control of the fuel pump stroke which controls the speed and power of the engine, each pump is under the control of an automatic safety stop which is mounted inside a case at the forward end of the pump shaft.

This safety stop which is operated by a small centrifugal governor is so arranged that when the engine reaches about 10 per cent. above normal speed, a lever comes into operation in conjunction with other smaller levers to lift all the suction valves on each individual pump thus putting them out of action in which condition they remain so long as the speed is above normal.

Fuel can be cut off from individual cylinders if desired by the hand operation on the smaller levers mentioned above.

This arrangment gives a very flexible job in so far as the engine can be turned over with only one cylinder firing it at any time it is required to do so.

One main fuel pipe supplies the pumps, but, of course, individual pipes carry the fuel from each pump to the individual cylinders.

The maneuvering gear on this engine is electrically oper-

ated and is so arranged that the operator only has to work one lever for maneuvering.

The gear is made up with three sets of levers which are synchronously arranged so that each movement of the operating gear is timed to take place at the correct moment and

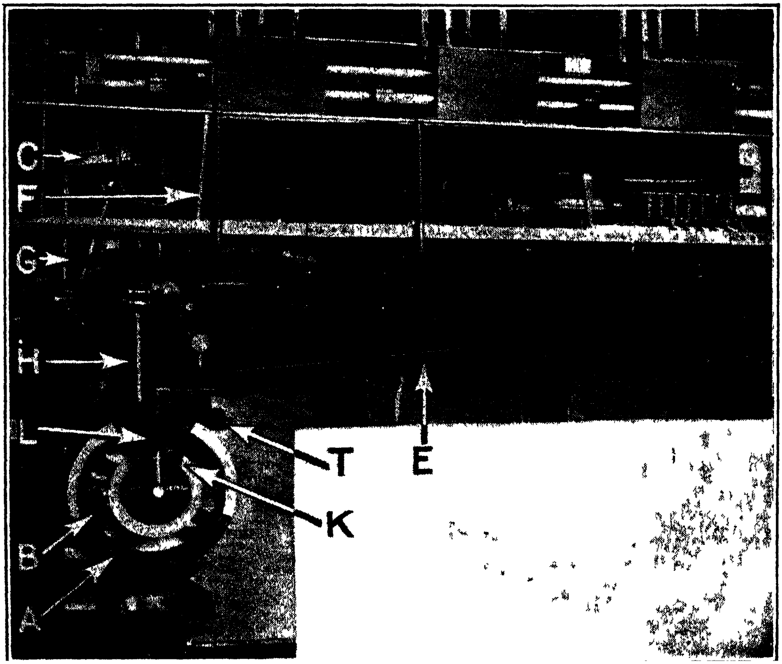


Fig. 206.—Control Gear of McIntosh and Seymour 2,500 H.P. Marine Engine

nothing can be done out of place or out of its proper time, see Fig. 206.

The lever "A" is the main operating lever, and projects horizontally from the face of the control ring. The pointer "B" is attached to the motor driving gears and follows the direction of the lever "A" at all times, whether the movement be for ahead or astern and is so arranged that when the lever "A" and the pointer "B" come directly in line with each other the electrical circuit is broken, thus stopping the motor, bringing the whole of the maneuvering gear to rest.

Should only a small movement of the mechanism be required, say, for changing the stroke of the fuel pump a very small amount, the small push buttons "K" are used. These push buttons close the circuit and the mechanism is in operation only while the man's fingers are on either of these buttons but the operation of these depends upon the position of the transfer switch "T".

The transfer switch "T" controls the current and passes same to the master switch to the push buttons as required.

The lamp "L" indicates whether there is current available or not for the operation of the mechanism.

With the lever "A" in starting position the motor, through a series of gears and levers and connecting rod "C" will cause the starting shaft located behind the camshaft to rotate, which places the air starting push rods in a position so as to be operated by the cams. Through this same series of levers a movement of the connecting rod "E" controls the speed of the engine. This connecting rod moves the driving shaft of the fuel pumps and changes the strokes of all the pumps at the same time.

For running ahead, the speed of the engine is controlled over the quadrant on the left hand side of the control ring and for running astern the speed of the engine is controlled on the quadrant on the right hand side of the control ring. To bring the engine from running ahead to running astern, the lever "A" is pushed downwards around the control ring to the astern quadrant and while the lever is passing through from the ahead quadrant to the astern quadrant the movement for running astern is taken care of as follows:

The connecting rod "F" controls the movement of a rocker shaft which removes the air, exhaust and fuel rollers from contact with their cams. While the connecting rod "G" through a bell crank moves the cam shaft endwise placing another set of cams under the cam rollers after which the rollers fall back to their original position for operating the valve levers.

The engine is then in a position for running astern and the compressed air enters the cylinders by means of the starting valves in the usual manner and the valve "H" automatic-

ally shuts off the injection air when the engine is being maneuvered.

A Kingsbury thrust bearing is fitted at the aft end of the engine and the casing for this is bolted on to the aft end of the engine bed plate. As will be seen from the illustration, photograph Fig. 207, this is made up as a complete unit so

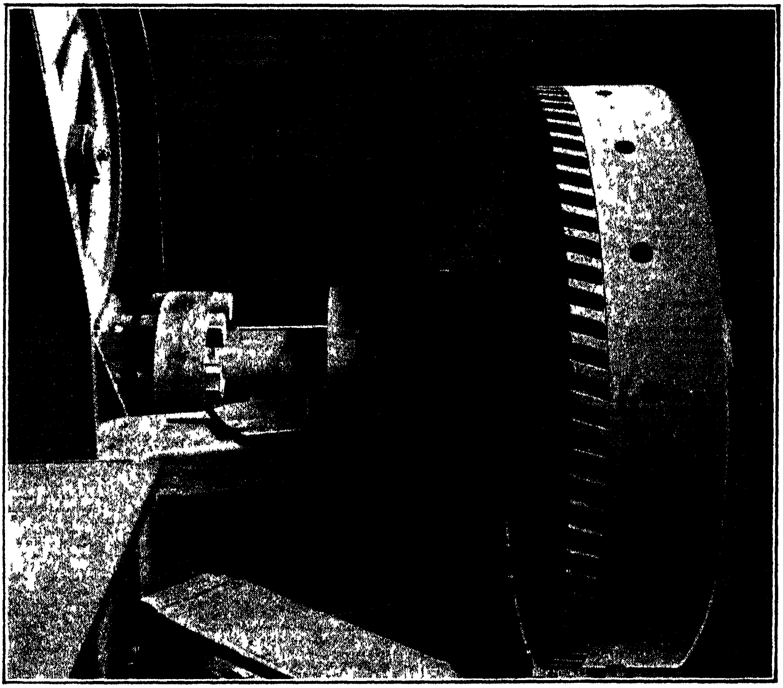


Fig. 207.—Kingsbury Thrust Assembly, McIntosh and Seymour 2,500 H.P. Marine Engine

that it can be shipped as a whole and thus avoid damage in reassembling in the ship. As is well known, the Kingsbury thrust bearing in its simplest form consists of one or more pivoted segments or shoes against which a thrust collar presses as it rotates. The bearing surfaces are well supplied with oil so that a perfect film is formed over these surfaces resulting in a very low co-efficient of friction. This bearing has been used on all of the smaller sizes of McIntosh &

Seymour marine engines and has proved very satisfactory and the results obtained in this case compare identically with the results obtained on the previous sizes.

Details of the bearing are shown on illustration Fig. 208. The principles of this bearing are so well known that it is not necessary to go into full details of them.

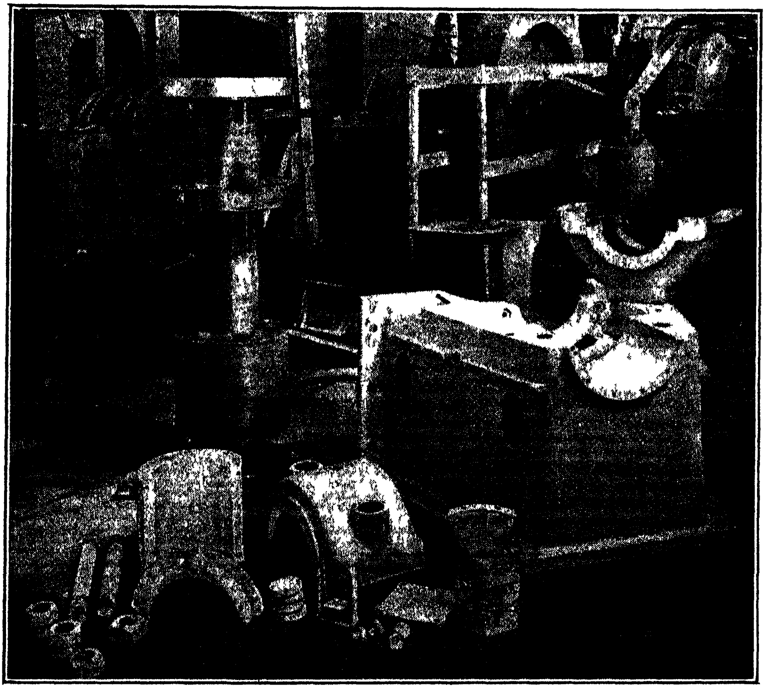


Fig. 208.—Kingsbury Thrust Details, McIntosh and Seymour 2,500 H.P. Marine Engine

The lubrication of the engine has been made to follow on the lines of previous designs of McIntosh & Seymour marine engines. The fundamental principle in the lubrication being to lubricate the pistons without any excess of oil. These engines are therefore arranged for forced lubrication, the oil being supplied under a pressure of approximately ten pounds per square inch into a main header pipe, from which it is passed through smaller pipes to annular grooves in the main

bearing. Each shaft journal is hollow and a radial hole permits the oil in this annual groove to pass through the middle of the shaft and through the crank webs and the crank pins to a radial hole in each crank pin. This hole is arranged to communicate with an annular groove in the crank pin box and the oil passes from this hole up through the connecting rod to the crosshead boxes. Another series of grooves in these crosshead boxes allow the oil to pass into the crosshead and through various openings on to the surface of the crosshead slipper, from whence it is returned to the base of the engine. Richard-Phoenix lubricators are employed for this purpose, which are driven by bevel gearing from the cam shaft. Another series of force feed lubricators supply oil by individual plungers to the main pistons through the cylinder walls and also to the compressor cylinders. The valve lever and push rod connections are arranged for hand lubrication, so as to insure that these parts are regularly inspected by the operator.

The engine base has a closed bottom so that all the oil that is fed to the working parts is caught and returned by means of a small pump back to the filter, where it is automatically filtered and passed back into the system. This arrangement of reuse of oil insures constant lubrication with a minimum expenditure of lubricating oil.

The Worthington Double Acting 2-Cycle Engine. A recent development of the 2-cycle double acting Diesel engine is that of the Worthington Pump and Machinery Corporation shown in Fig. 209.

Sound standard practice is followed in the design of the crankshaft, connecting rod and frame. The frame is of cast iron, cast in box section. It is entirely enclosed with oil-tight doors suitable for pressure lubrication. The crank case is, however, entirely separated from the cylinder, so that no cylinder gas can escape into it, even if the piston rod stuffing box should leak, and lubricating oil cannot make its way from the crank case into the cylinder.

The cylinder base, also of cast iron, rests upon the frame, and to all practical intents forms part of it. It is, however, bolted directly to the bedplate by four tie bolts, which take the tension stresses. It contains the passages for scavenging air

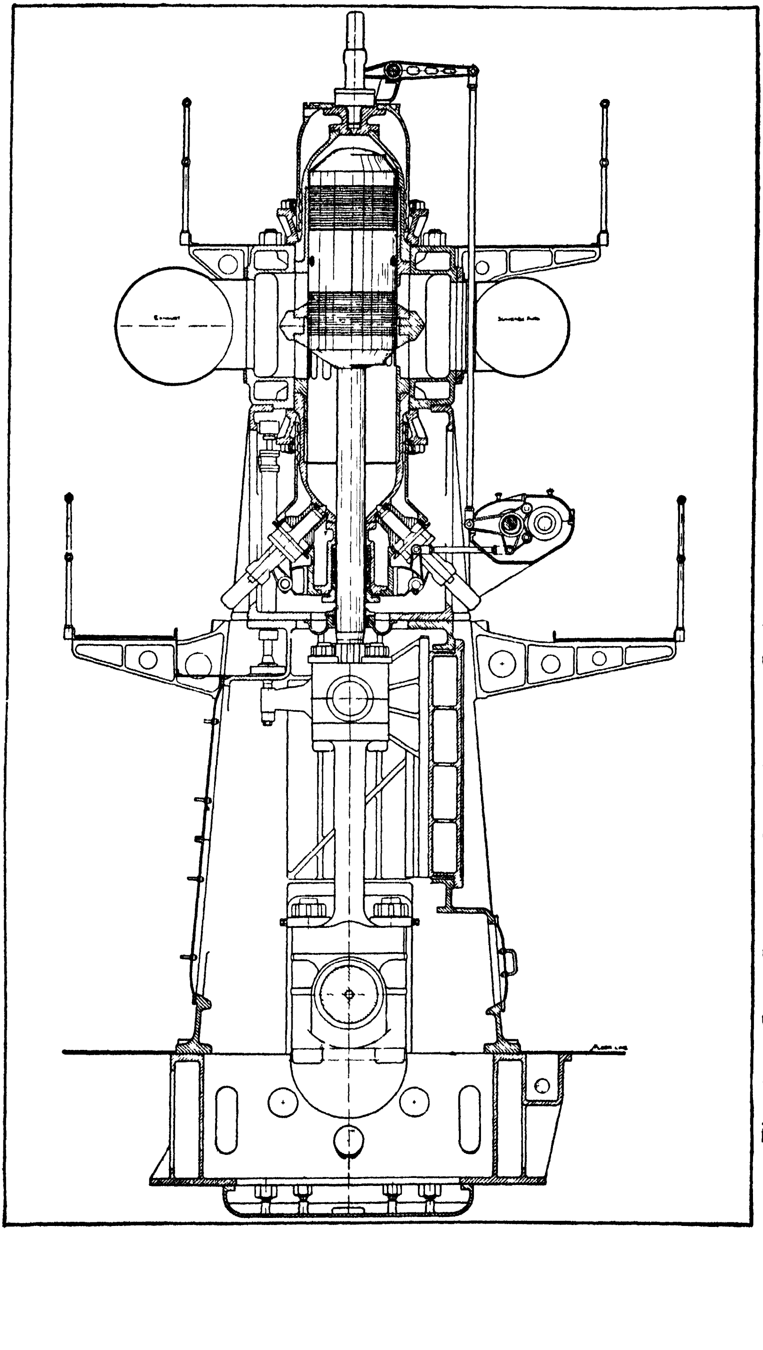


Fig. 209.—Cross-Section of Worthington 2-Cycle, Double Acting, Air Injection Marine Engine

and exhaust, and for the cooling water circulation, for both top and bottom cylinders.

The top (Fig. 210) and bottom cylinders are alike except for the piston rod stuffing box which in the bottom cylinder occupies the corresponding point to the top fuel supply valve; and the two bottom fuel supply valves, which are placed symmetrically opposite each other, entering the cylinder at an angle, just above the stuffing box. The design is worked out

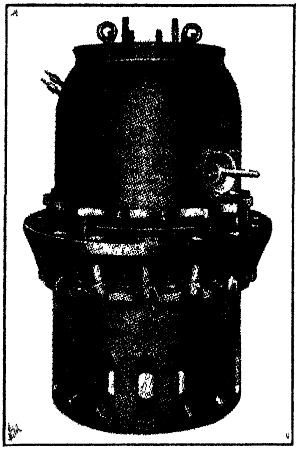


Fig. 210.—Upper Cylinder, Jacket and Clamping Ring Removed from Base, Worthington Double-Acting Engine

to secure a uniform and symmetrical distribution of the fuel charge in the bottom cylinder as well as in the top.

Each cylinder consists of a steel shell so shaped as to secure the required strength for the internal pressure, with the least thickness of metal. This permits higher heat generation rates than is possible with cast iron without injury to the metal. There is no cylinder head joint as the head and half cylinder are one piece of steel, and this also contributes to safe working at high capacities.

There is a substantial shoulder on the outer circumference of the half cylinder, by which it is secured to the cylinder base by means of a clamping ring, which in turn is bolted to

the base by heavy studs. This leaves each cylinder half with its integral head freed to expand as it becomes warm. Inside the steel shell is pressed a cast iron liner, which contains, at the open end, the ports for scavenging air and for exhaust of the gases. The free end of the liner rests on a heavy flange projecting from the inside of the center bore of the cylinder base, the upper and lower cylinders thus being held in perfect alignment by the bore.

The cooling water circulation is from the center toward each end of the cylinder, being introduced in the center of the cylinder base and flowing up through the top cylinder jacket and down through the bottom cylinder jacket, regulated at the outlets by two independent valves. The two cylinders are entirely free to expand independently from the clamping ring toward the outer end, where combustion takes place and the metal temperature is consequently greatest. The scavenging and exhaust ports lead off independently from the common passages in the cylinder base, to the respective ends.

The valve motion is designed on conventional lines with one cam for each fuel spray valve. Each valve lever oscillates round an eccentric on the fulcrum shaft, which can be turned through a small angle by hand for reduction of the valve lift at low speed. The cams are symmetrical and can be used for running in both directions by rotating them through an angle of 34 degrees in relation to the main shaft. This is done in the following manner:

A spur gear on the main shaft engages an idler of considerable width, which drives a spur gear wheel keyed on a horizontal lay shaft located in the bedplate in front of the engine. This shaft can slide longitudinally in its bearings and, as the above mentioned spur gears have diagonal teeth, this motion will cause it to rotate through an angle of 34 degrees for the full travel of the lay shaft. The cam shaft is driven from a bevel gear bolted to the sleeve by the intermediate vertical shaft and will thereby be displaced the proper angle when the lay shaft slides from one extreme position to the other. To the forward end of the lay shaft is attached, by means of two ball bearings, a piston in the bore of which the lay shaft can freely revolve. This piston travels in a cylinder under oil

pressure and is held tight against a face in the one or the other end of the cylinder for either ahead or astern rotation of the engine. Reversing is thus reduced to the turning of a four-way cock, admitting oil on one side and allowing oil to escape from the other side of the piston.

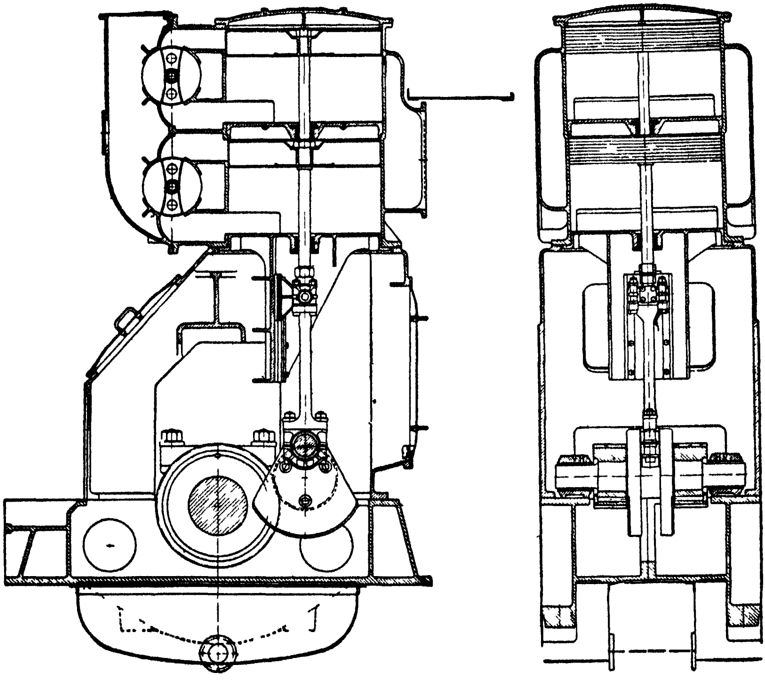


Fig. 211.—Scavenging Pump of Worthington Double-Acting Marine Engine

The scavenging operation is accomplished on the low pressure, large volume principle. A single scavenging pump, Fig. 211, serves for a four-cylinder engine, being driven by a crank and crosshead connection. The scavenging pump crosshead also acts as the first stage of the spray air compressor, the other three stages being located on the rear of the engine, but driven by the same connection.

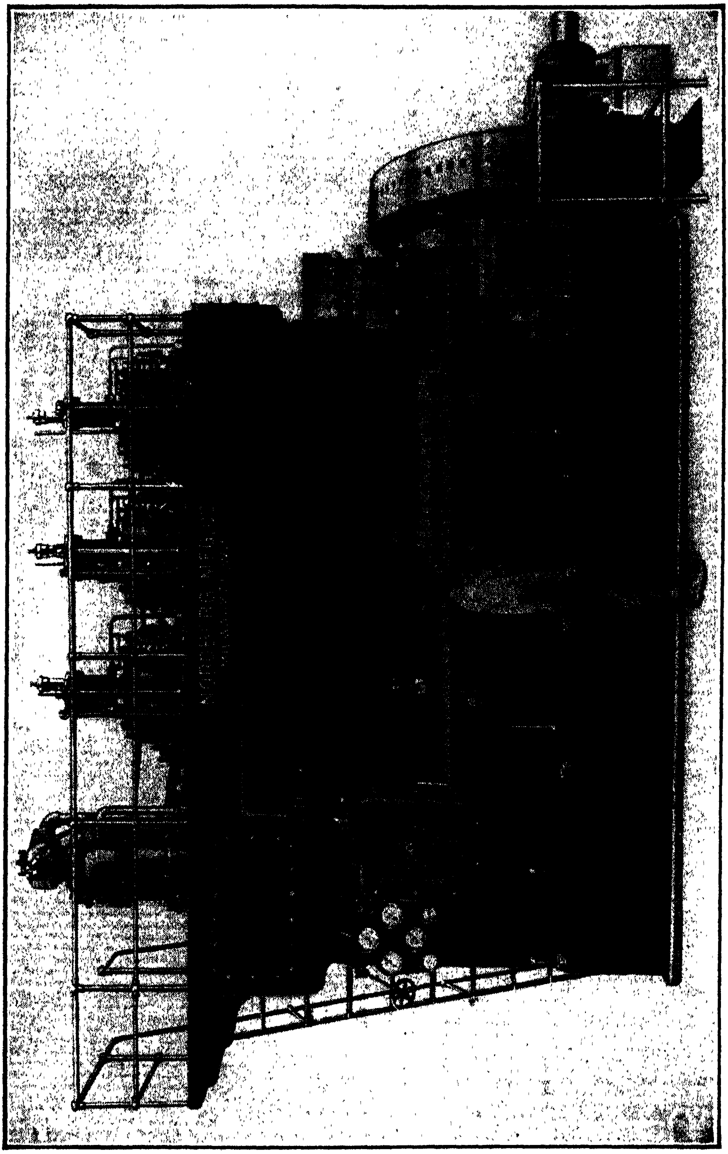


Fig. 212.—Nordberg Type V.H. Three-Cylinder, 2-Cycle, Air Injection Stationary Engine

During a recent thirty-day non-stop full power endurance test of a single cylinder engine of this type the following results were obtained:

Duration of test - - - - - - - - -	720 hours
Total number of revolutions during test - -	3,879,921
Average speed - - - - - - - - - -	89.8 R. P. M.
Average brake load - - - - - - - -	2,400 pounds
Length of brake arm - - - - - - - -	15 feet
Average brake horsepower - - - - - - -	615
Average indicated M.E.P. top cylinder - - -	81 pounds
Average indicated M.E.P. bottom end - - -	77 pounds
Average indicated horsepower - - - - - -	778
Fuel oil consumption average of four two-four tests per brake horsepower hour - - - - -	0.428 pounds
Gravity of fuel used - - - - - - - -	28 to 32° Baumé
Fuel oil consumption per I.H.P. per hour - -	0.339 pounds
Mechanical efficiency measured during the four tests - - - - - - - - - - -	79.5%
Average lubricating oil consumed in the cylinder and piston rod stuffing box per 24 hours - - -	4.03 gals
Average exhaust temperature - - - - -	510° Fahr.

The Nordberg Type VH Engine. A new design known as the Type VH engine has been added to the line of Nordberg Diesels to give a complete range of medium-sized units. Operation is on the full Diesel principle involving the injection of fuel by means of a blast of highly compressed air. The engine is of the 2-cycle, single acting, vertical type built in three, four, five and six cylinder units, ranging in size from 450 to 900 H.P. A cylinder size of 16¾ x 22 inches gives an even rating of 150 H.P. per cylinder at a speed of 225 R.P.M. A 3-cylinder unit is shown in Fig. 212.

From previous designs a number of variations have been made to simplify construction and some new features incorporated, such as forced feed lubrication of the bearings, oil cooling of the pistons, the use of cross-heads in a comparatively small engine, dual wiper rings on the piston to separate cylinder and bearing oils and what is more apparent in an external view, a box frame enclosing the working parts.

The engine is made up of three principal castings consisting of the bed-plate in one piece, the frame and the cylinder bloc, all held together by long through bolts extending from the bed plate to the top of the cylinders. The cam shaft bracket is cast integral with the frame and the scavenging air manifold integral with the cylinders. The upper side of this manifold

has been made flat to serve as a platform, giving access to the working parts above.

Oil, rather than water, is used for cooling the pistons, the supply coming from the pressure lubricating system, which is of sufficient capacity to serve the twofold purpose. Draining

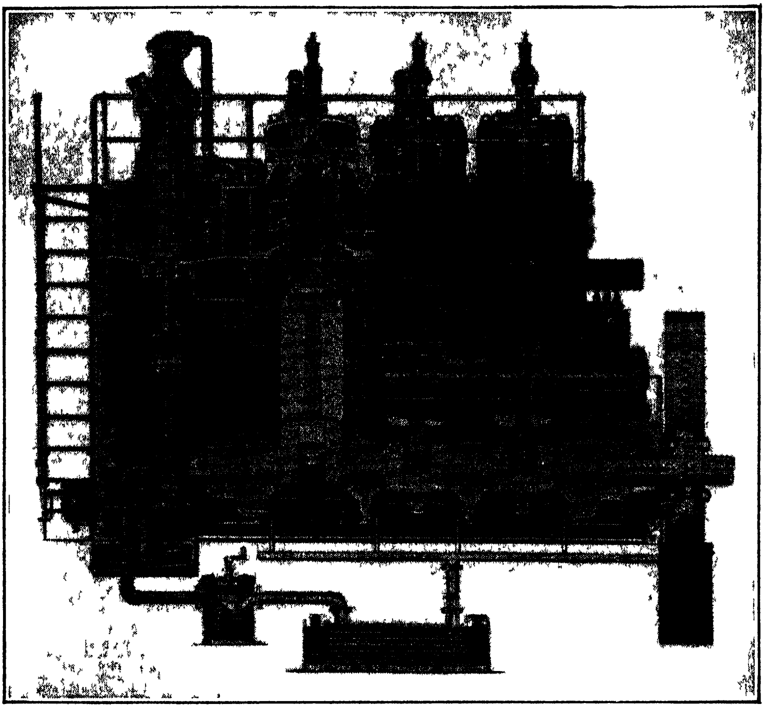


Fig. 213.—Sectional View of Nordberg Type V.H. Engine

back from the bearings into the crank case, the oil collects in a sump at the air compressor end of the unit, as shown in Fig. 213. A rotary pump, gear driven from the crankshaft, forces the oil through a twin strainer and a tubular cooler through which all cooling water for the engine is passed, to a pressure header, from which it is distributed to the several main bearings. Continuing through radical holes into a hollow crankshaft, the oil is distributed to the crankpins and cross-

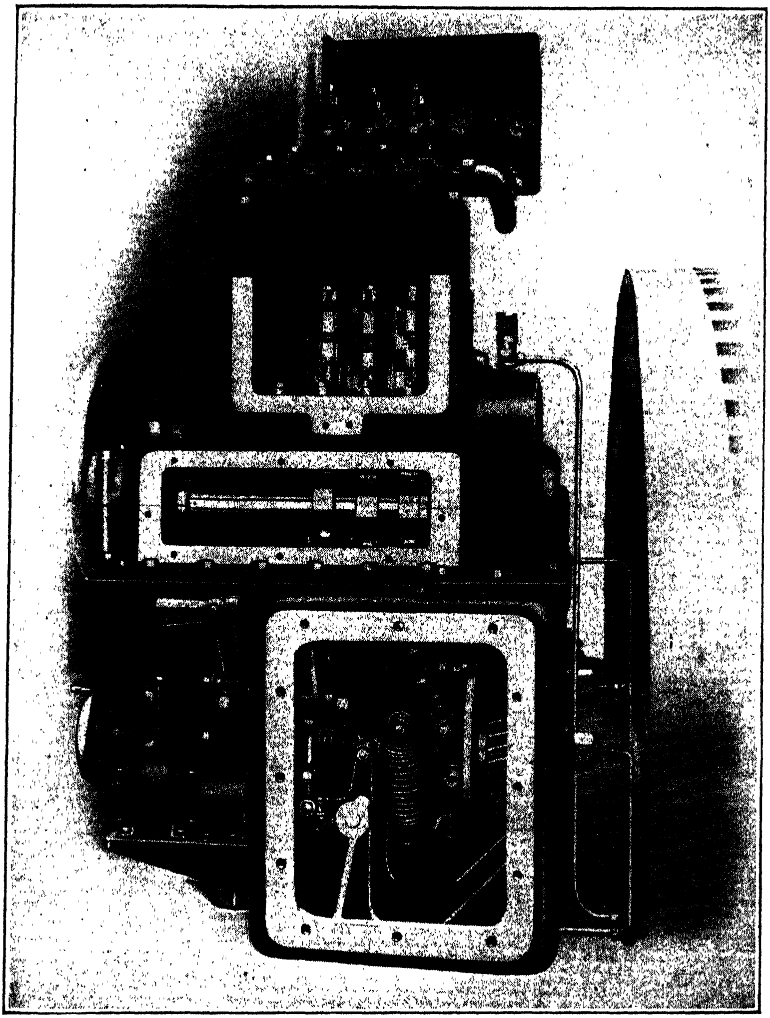


Fig. 214.—Fuel Pump and Governor, Nordberg Type V.H. Engine

heads and from the latter passes through a special pipe connection to the piston head, where it flows through a continuous spiral passage and returns to the crosshead through a pipe connection on the opposite side, from which it passes out into an open spill pipe leading to the oil sump. At the pump a

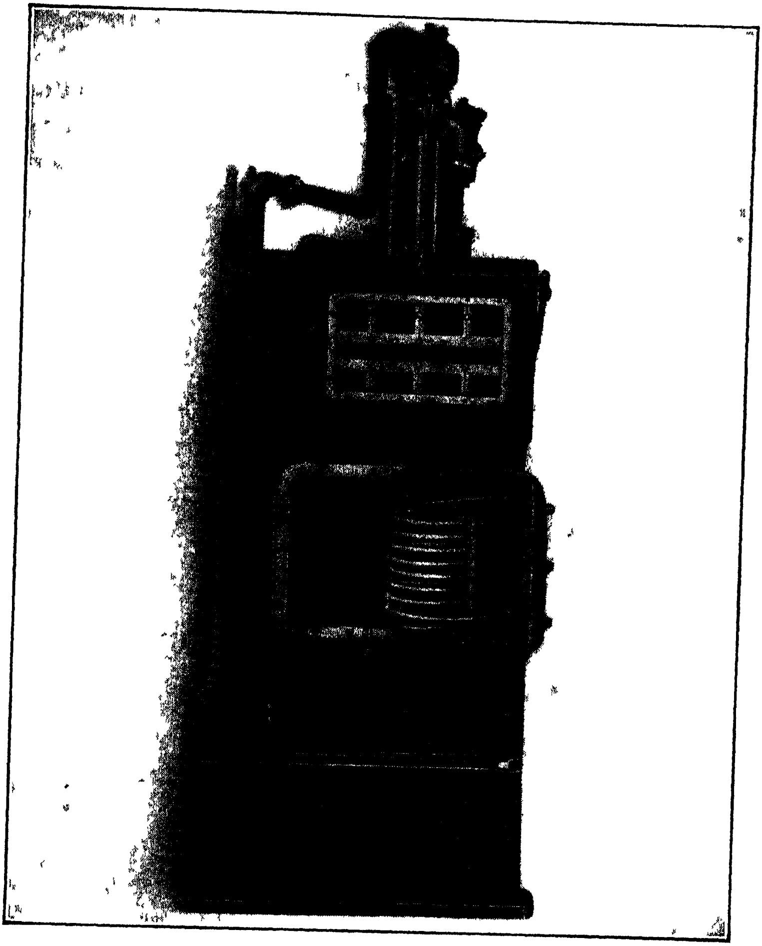


Fig. 215.—Air Compressor and Intercoolers, Nordberg Type V.H. Engine

spring-loaded spill valve, operated manually, is provided to regulate the oil pressure in accordance with the load on the engine. From the pressure header connections are made to lubricate the bearings of the camshaft, governor and fuel pump. Lubrication of the power and air compressor cylinders is effected by force-feed lubricators driven from the camshaft.

In connection with the lubricating system dual wiper rings at the bottom of each cylinder prevent the contamination of pure oil by the foul oil coming from the cylinder. The upper ring catches the foul oil which is collected and carried to the outside, while the bottom ring wipes off the surplus clean oil

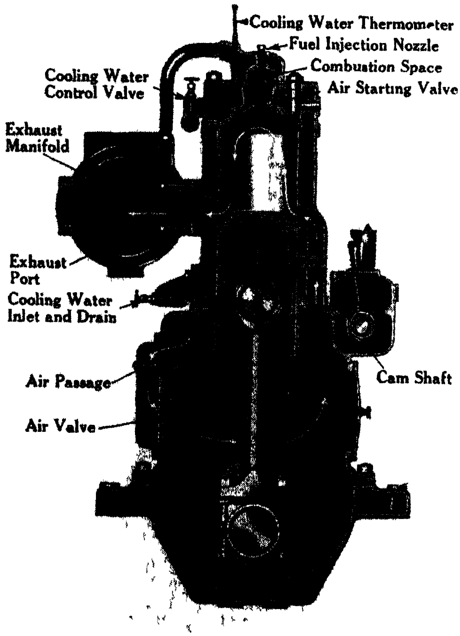


Fig. 216.—Transverse Section Through Fairbanks-Morse Marine Engine

coming from the crankpit and returns it to the lubricating system.

The fuel pump, which is of the constant-stroke type, is built up into a complete unit and mounted on top of the governor housing. The body of the pump is made from a forged-steel billet machined to receive the plungers, one for each cyl-

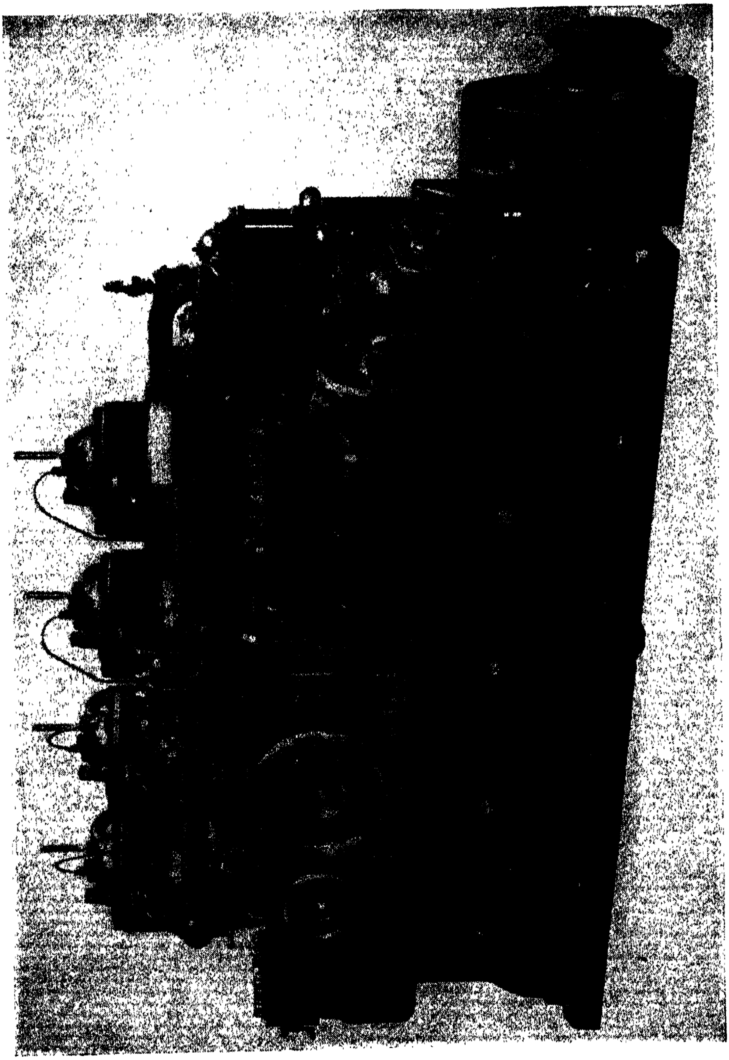


Fig. 217.—Front View of Fairbanks-Morse Marine Engine

inder, and also the suction and discharge valves which are mounted in cages. The plungers are operated by cams on a camshaft passing through the fuel pump housing and driven by spiral gears from the crankshaft. For priming each pump plunger can be operated by hand.

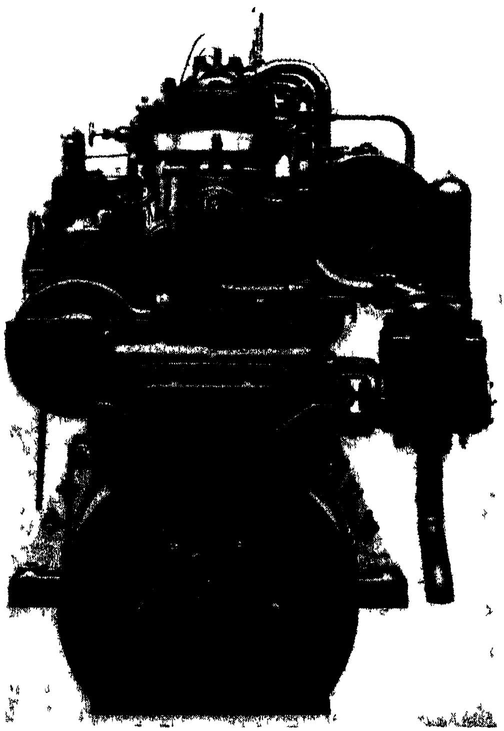


Fig. 218.—End View of Fairbanks-Morse Marine Engine

The governor functions by acting on the suction valves of the fuel pump, closing them earlier or later, depending upon the load and the amount of fuel required. Fig. 214 shows the entire fuel pump unit, the governor and a small motor-driven synchronizing device provided to permit speed control from the switchboard for parallel operation with other power units.

As in previous designs by the same company, the air compressor and scavenging pump are combined into a single unit

at the end of the power cylinder and driven from the main crankshaft. The scavenging pump is below, and above is a three-stage compressor to furnish high-pressure air for starting and fuel injection. The former is a double-acting low-pressure pump fitted with strip valves to deliver air to the scavenging header at three-pound pressure.

For the air compressor copper-coil intercoolers, Fig. 215, have been located in cored cavities in the frame, there being an intercooler between the low and intermediate and high and an aftercooler placed beyond the high-pressure stage. The coils are attached to the covers of these cavities, which have been designed as water separators to remove moisture entrained with the air. The coils are attached to the covers so that the separator and cooler can be removed as a unit.

The New Fairbanks-Morse Marine Diesel Engine. The Fairbanks-Morse Company have recently redesigned their type C-O marine oil engine into the airless injection type of Diesel engine. The new design is not, however, a radical departure from the well-known C-O engine of which over 150,000 horse power are in successful operation. The principal improvements are those which adapt the engine to the use of a wider range of low grade fuels, better fuel economy and means for immediate starting without the aid of auxiliary ignition devices. A transverse section through the cylinder of this engine is shown in Fig. 216, a front view of the engine in Fig. 217, and an end view in Fig. 218.

Two of the first points to strike the observer in the new design are the compactness and simplicity of the engine. While fundamentally due to the 2-cycle principle and to the use of airless injection, the simplicity is chiefly the result of careful design. All piping has been eliminated or closed, and the starting, reversing and governing mechanism has been worked out in a compact unit located at the center of the engine. The goal of the designers was to produce an engine of maximum simplicity and reliability, and that object has been achieved by thoroughness in the execution of the details. From end to end of the engine there is evidence of careful study of every detail.

An interesting feature of the Fairbank's Morse design is

the use of what the engine builders term 2-stage combustion, the arrangement of which was worked out at the Beloit plant after extensive experimental research had demonstrated its many advantages.

A few degrees before the compression has been completed, i.e., near the top dead center of the piston travel, a spray of fuel is introduced into the combustion chamber through a nozzle centrally located at the top. Partial combustion, which results in burning the fuel charge in the pre-combustion chamber to carbon monoxide (CO) with the liberation of a relatively small amount of heat, takes place as the first stage of the process, and the liquid fuel that has entered the cylinder combustion space is gasified. Because of the partial nature of the first stage of combustion there is no pressure rise of any consequence.

As the piston starts on the downward stroke, with the attendant rush of gases through the neck into the cylinder space, a considerable degree of turbulence and a thorough mixing of the air excess with the CO gas and oil vapor are brought about and the final combustion of the charge to carbon dioxide (CO_2) is completed.

Over 50 per cent of the heat due to the entire combustion is liberated in the second stage, but owing to the fact that the piston is then beginning to pick up considerable speed the expansion quite accurately neutralizes any rise in pressure which would otherwise result from the more intensive and complete 2nd stage combustion. So thoroughly has the system been worked out that flat top indicator cards are obtained, and the engine works with spontaneous ignition of fuel and complete burning unaccompanied by pressure rise, according to the method first conceived by Diesel.

An important advantage of the 2-stage combustion scheme is that its success is not dependent on the use of fine multiple orifices in the fuel injection nozzles or the use of high injection pressures. An interesting feature is the approximate equalization between the velocities of the oil jet and of the air entering through the neck toward the end of compression. It was established that an insufficient velocity did not give the best results because of deficient turbulence and that an

excess of velocity had a tendency to throw the fuel back against the cooler portions of the combustion chamber. Careful design and precision in manufacturing methods have made it possible to secure the proper balance between air and jet velocities.

A careful distinction is to be made between the system here used and those pre-combustion methods which depend upon an actual explosion for projecting a part of a liquid fuel charge into the cylinder.

From the practical operator's point of view, the use of a low velocity fuel oil jet has the important implications that not only are excessive fuel pump pressures avoided, but it is possible to employ unusually large atomizers and injection nozzle diameters. Actually the nozzle diameter is 5/64 in., and the helical stem placed just ahead of the nozzle has course grooves of nearly the same size. A spring-loaded check valve placed inside the body of the nozzle makes for a sharp beginning and ending of the jet. In view of the large passages through it, there would hardly seem to be any occasion for ever taking it out; nevertheless, it is accessibly mounted and can be removed in less than a minute by simply unscrewing it.

Complete control of the engine is centered in a compact and accessible unit located at the center of the engine and housing the injection pumps and air starting and reversing mechanism. A governor unit with its handwheel for speed control is mounted alongside the injection and air starting unit, thus centralizing the complete control system.

By referring to the diagrams of the control unit and the governor, Figs. 219, 220, 221 it is possible to appreciate the simplicity of the general control scheme. When the control handwheel for starting the engine is turned, the main air valve is opened by a cam on the control shaft. The compressed air passes through this master valve to a passageway which supplies air to the individual valves for each cylinder. The air starting cams revolve with the camshaft, raising these individual valves and admitting air to each cylinder in the firing order.

During the admission of compressed air, the fuel control cam turns an arm underneath it, which in turn moves a shaft

raising the suction valve rockers so that fuel cannot be injected. When the engine begins to turn over on compressed air, the control wheel is turned back to the running position and the fuel control cam releases the suction valve rockers, the

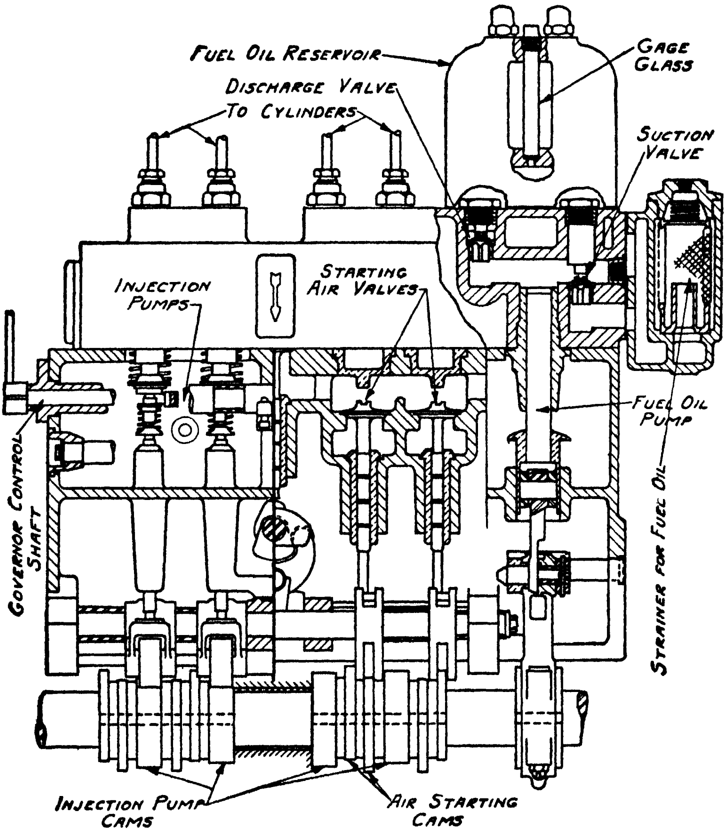


Fig. 219.—Longitudinal Section Through Fuel Pump and Starting Valve Unit, Fairbanks-Morse Marine Engine

fuel pumps then being enabled to inject fuel to the cylinders and firing commences.

When the control wheel is turned from the start to the "run" position, the main air valve is closed and a relief valve is opened, allowing the air pressure in the space leading to the master valve to escape to the atmosphere.

It will be noted that the camshaft also carries the cams which operate the fuel injection pumps. As the cam revolves, it raises the injection pump roller which in turn actuates the injection pump plunger. It is not possible, however, for fuel

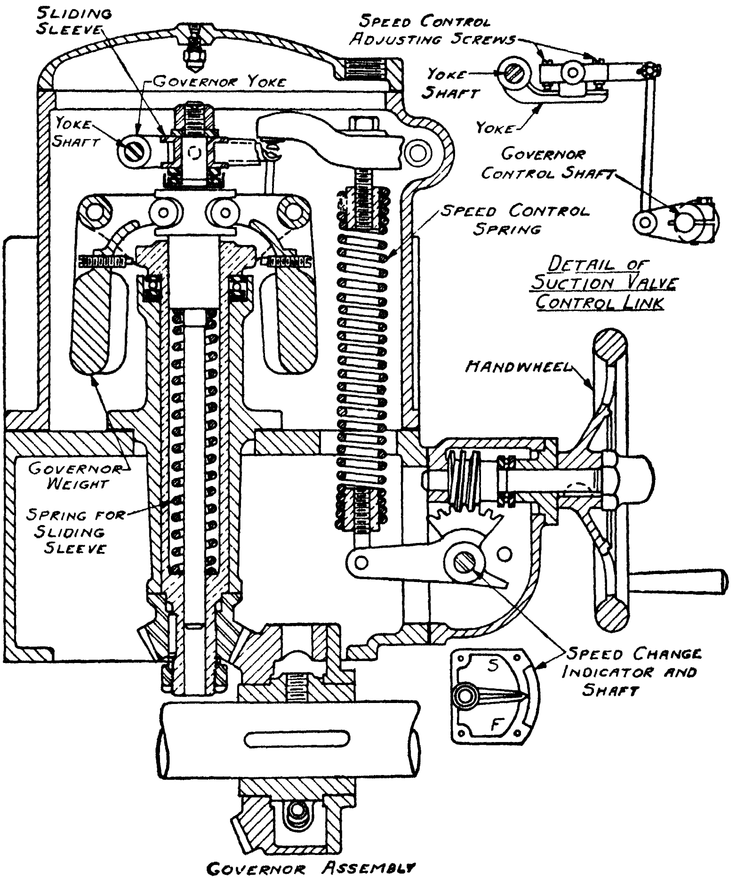


Fig. 220.—Section Through Governor Unit, Showing Speed Control Mechanism, Fairbanks-Morse Marine Engine

to be injected through the discharge valve to the cylinder until the suction valve has closed. The closing of this suction valve is accomplished by means of a valve rocker actuated by the upward movement of the injection pump plunger. This suction valve rocker is pivoted on a shaft that is turned

through a slight angle as the pump plunger moves upward. As the rocker drops, the suction valve closes and the fuel oil

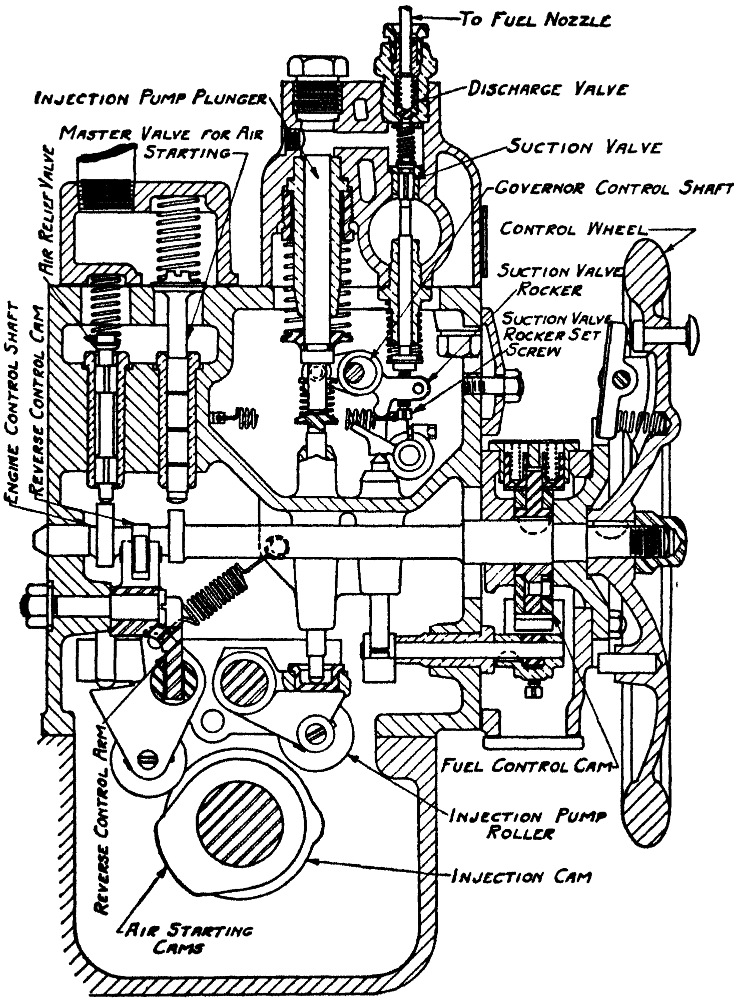


Fig. 221.—Cross-Section of Control Mechanism Unit, Showing Master Cams, Fairbanks-Morse Marine Engine

is trapped so that its only course is through the discharge valve and to the cylinder.

Since the amount of fuel injection is dependent on the load carried by the engine, the suction valves have to be placed

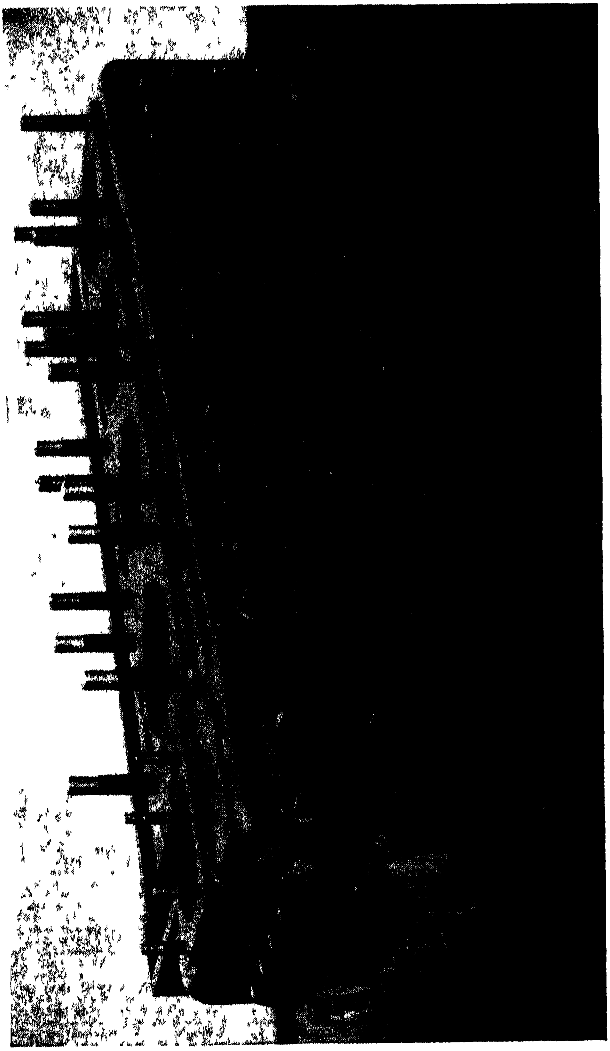


Fig. 222.—Upper Base Casting of the Fairbanks-Morse Marine Engine Seen from the Forward End at the Back

under governor control. The suction valve rocker is pivoted in eccentric bearings and the amount of rotation is determined by the position of the governor weights linked with this shaft as indicated on the sectional view of the governor. When the load on the engine is increasing, the suction valve closes earlier in the stroke of the pump plunger, and thus more fuel is injected.

For reversing the direction of rotation of the engine, a cam on the engine control shaft actuates a roller pinned to the reverse control arm. This arm in turn moves the air-start rockers and rollers along the shaft until the rollers are directly over the correct cam. When the air is then admitted to the individual air-start valves, these push the air-start rockers with their respective rollers down on the cams, and the position of the cams determines which valves remain open to start the engine in the desired direction.

The governor is of the flyball type, having a vertical spindle driven from the camshaft by a flexible drive bevel gear. As the engine speed increases or decreases, the weights swing at a varying radius which causes the sleeve to slide up or down the spindle. The governor yoke is keyed to a shaft which can turn slightly, and the movement of the shaft is transmitted to the fuel pump suction valve rocker shaft as shown in the diagram. The adjusting screws are set for the correct cut-off of fuel at the rated horsepower and speed. Thus the governor controls the amount of fuel injected by regulating the effective length of the injection plunger stroke.

A large range of speed variation can be obtained by turning the speed control wheel. A reduction of speed is obtained by putting tension on the spring, which in turn pulls down the arm to which it is held at the top, and this arm forces down the control rod. The control rod turns the fuel pump suction valve rocker shaft and thus has the same effect as if the governor weights were thrown apart. It holds open the fuel pump suction valves for a longer period of the injection stroke and decreases the amount of fuel injected.

On referring to the transverse section one can note that the engine is built up of three main sections, the lower base, the upper base and the individual cylinders. The lower base

is a single casting of heavy construction provided with wide box-section flanges on both sides for bolting to the foundation. These flanges are located just below the center line of the shaft. The lower base forms the lower half of the crank case, and heavy bridges support the main bearings and at the same time separate the crankcase compartments.

The upper base, Fig. 222, is also made in a single casting of sturdy cross section. Large doors open into each crank case affording access to the main and crankpin bearings. As in the case of the lower base, ribs are used in the upper base to separate the crank case compartments with the exception of a clearance space above each bearing to allow for its removal. This clearance is covered by a sheet metal disc bolted to the bearings and the upper base, making an effective air seal.

A compartment is cast in the exhaust side of the upper base, and this passage runs the full length of it to serve as an air duct for the scavenging air. From this air duct the air is drawn into the crank case through automatic grid valves as the piston moves upward and creates the suction. When the piston returns on the down stroke and uncovers the air inlet port of the cylinder, this air, which has been trapped in the crank case, passes through the port, is deflected upward by the piston and completely scavenges the cylinder of any products of combustion which remain.

A crankshaft conforming to the highest Diesel engine specifications is bedded in long journals and works under lower specific bearing pressures than in the old engine in spite of the fact that the compression in the newer type has been raised to 500 lb. per sq. in. A high degree of precision in the machining of the crankpins is obtained by the use of a special crankshaft lathe, in which the crank itself is held stationary while the tool revolves around the pin. Corresponding to the ring lubrication of the main journals, a banjo oiler for the crankpins is used, and by the clever expedient of mounting it eccentrically in the direction of the crank radius the oil is supplied to the pin with an extra amount of centrifugal force.

Both the piston and the cylinder, Fig. 223, in which it works are cast of a special grade of iron containing approximately 20 per cent of steel and possessed of extra wearing qualities.

Compression relief is arranged for by providing a small valve in each cylinder head. A steel cage is screwed into the

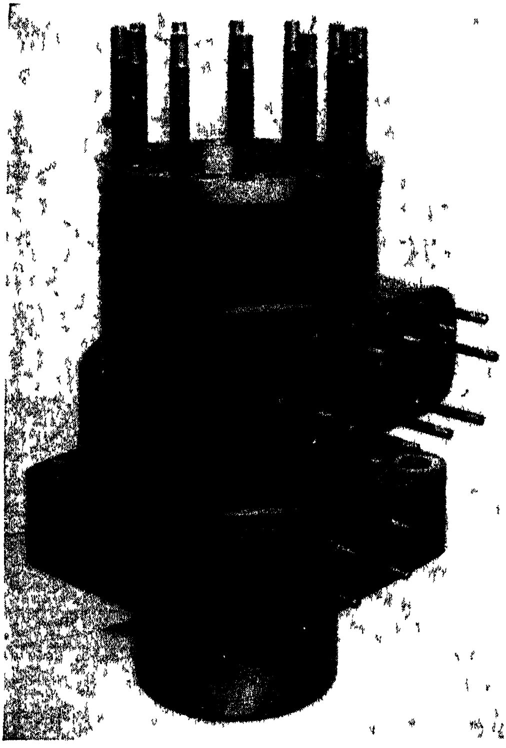


Fig. 223.—Cylinder Casting of Fairbanks-Morse Marine Engine, Showing Exhaust Ports

cylinder head and contains a spring loaded valve held on its seat by compression on the cylinder during operation. These valves are operated by a layshaft which in turn is manually actuated by a suitable gear.

In this engine the piston, Fig. 224, has been made of a length characteristic of conservative marine Diesel engine practice, and the ratio of effective length to diameter exceeds

2:1. All longitudinal ribbing is avoided and stiffness is imparted to the piston structure by circumferential webbing

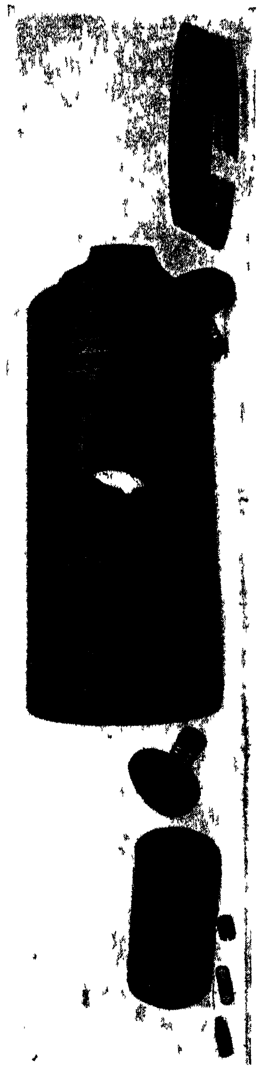


Fig. 224.—Piston, Drilled Wristpin, Oil Retainers and Piston Rings, Fairbanks-Morse Marine Engine

only. It has been found that lengthwise ribs sometimes have a tendency to throw the piston barrel out of round, and the fact that only circular webs are used is one more indication

of the fact that in this design details have been carefully looked after.

One of the most interesting lessons in the design of Diesel engines may be illustrated by reference to the upper end of the piston used in this particular machine. It will be seen that a crown of substantial thickness is used, not so much for the sake of strength as for providing an adequate section for the transmission of heat. It will be noticed also that there is a gradual taper in the thickness of the piston section from the under side of the crown to the interior of the barrel, and the extra metal at this point further carries out the idea of providing for a free flow of the heat.

It has been found that the side surfaces of piston rings are one of the most effective means for getting the heat of the piston away to the walls of the cylinder and thence to the jacket, a consideration which has been borne in mind in providing this engine with six piston rings. They not only serve as a means for sealing the compression, but also have an important bearing on the matter of keeping the piston crown cool.

Inside of the piston a deflector is provided for preventing lubricating oil from being splashed against the crown. It amounts hardly to more than a deepening of one of the circumferential webs above the head of the rod, but it is so arranged that it fully intercepts any splash of oil which might otherwise lodge on the under side of the piston crown and carbonize there. It is hardly necessary to point out that, in the absence of provisions of this kind, carbon crusts of considerable thickness are apt to be formed, and when they crack loose and fall down they may interfere with the lubricating system. In some designs this difficulty has been obviated by bolting a cover plate inside the piston above the end of the rod, but it would seem as though the bolted cover would shut off a free circulation of air from underneath the crown and might possibly prevent as effective a cooling of the latter as if it were left exposed to the free circulation of air.

A die-forged connecting rod—one of the largest of this kind manufactured—is fitted with a bronze babbitt-lined crank-pin box in two halves and is permanently aligned to the latter

by means of a longitudinal key. Straight-shank bolts are used for fastening the box halves together and to the foot of the rod, and as the key takes care of the alignment there is no need for having the bolts body-fitted. Interchangeability in manufacture and replacement is therefore fully preserved. At the piston pin end of the rod a shell bearing is used, insuring against cramping consequent on heat expansion by the fact that it has a saw cut near the top of the shell.

During the scraping of both the connecting rod bearings, micrometer fixtures are used for checking the parallelism of the two. In consequence of the delicate measuring apparatus used, it is impossible for the two bearings to be out of parallel either in the vertical or horizontal plane.

The piston transmits its motion to the connecting rod through a hardened piston pin of large diameter. Here again the amount of bearing surface has been increased so that the pressure is not materially different from that found in the former design.

For the supply of lubricating oil to the piston pin a movable side scraper, working in a counterbore of the pin, is used, and at assembly it is carefully bedded to the radius of the cylinder against which it is urged by means of a light spring. A Vee-shaped recess cut into the cylindrical contour of the scraper and terminating in a throat at the base of the Vee-surface is an effective lubricating oil collector. Leading from the counterbore in the end of the piston pin is a steel pipe with a drop nozzle securely brazed into place, a provision which has eliminated the possibility of its loosening or breaking off in service.

One of the features of the piston pin lubricating arrangement is the dependence on a large number of holes through the walls of the hollow pin for the proper distribution of the oil over the bearing surface. The pin, of course, is pack-hardened and ground, but before the hardening process is carried out, the lubricating holes have been countersunk with radii, obviating all sharp edges and providing for the positive formation of the lubricating oil film.

An ordinary taper dowel pin is used for locking the piston pin into the piston bosses, only one of which is a tight fit on

the pin. A spring behind the taper dowel insures that it will effectively lock the pin both against rotation and sidewise movement, and at the same time the ends of the spring bear on offsets in the ends of the dowel and of the threaded spring

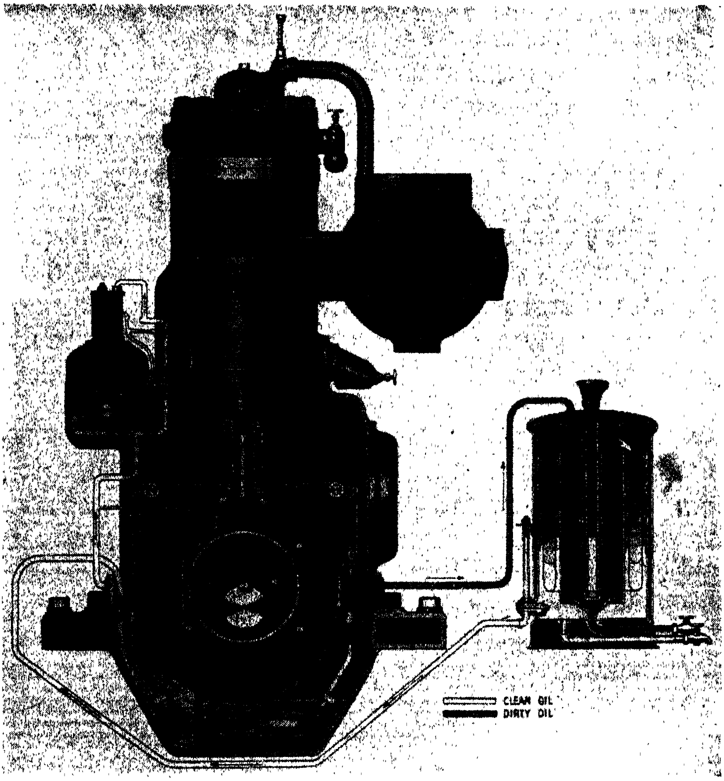


Fig. 225.—Diagram of Oil Pumps, Filter, Feed Tank and Sump, Fairbanks-Morse Marine Engine

retaining plug in such a way that the possibility of the plug backing out is obviated.

Owing to the fact that possible shocks, and a consequent tendency to "work," are not transmitted from the dowel pin by means of the spring the device forms a positive lock.

In order to still further simplify the operation of the engine and to assure the maximum reliability of service, the lubri-

cating system has been made completely automatic. That this has been thoroughly worked out is shown by the fact that there is not a single place in the engine that requires manual lubrication. There are no oil holes or grease cups to be found anywhere on the engine.

At the aft end of the base Fig. 225 is a clean oil reservoir in which the suction of the distributing oil pump is set. A similar pump is placed at the other side and its suction is set in the sump at the bottom of the base. Both pumps are identical and are driven by means of arms and links. Oil is forced to the cylinder, piston pin, crankpin and main bearings by force feed lubricators as shown in Fig. 226. From these bearings a surplus finds its way to the lower crank case and is drained to the sump. The dirty oil is then pumped to the oil filter and after being cleaned is fed to the sump chamber for clean oil, from which it is pumped into the lubricator and camshaft oil pipe header.

The camshaft bearings, cams and eccentrics are lubricated by means of the camshaft oil pipe header, to which oil tubes and oil spitters are connected. Clean oil is pumped directly to the oil pipe header by means of the clean oil pump. This same pump also supplies clean oil to the lubricators for distribution. The excess oil pumped to the lubricators overflows into the sump chamber.

In order to warm the oil so that it will filter freely in cold weather the filter is provided with a water jacket which can be piped up to the hot water discharge from the circulating system. The filter is equipped with a 4-section screen which removes any impurities accumulated during the passage of the oil through the engine.

At the aft end of the engine an eccentric drive is provided for the air compressor and the circulating pump. The air compressor is of ample size to meet the usual conditions. It is single-acting with a trunk piston. Spring loaded poppet valves are used at the air inlets, while the discharge valves are of the cup type, spring loaded. An unloading device is located in the cylinder head and operated by pulling a knob to open the suction valve.

The eccentric sleeve which drives the compressor is lined

with babbitt and the bearing surfaces are machined on a spherical radius from the center, so that no binding will be caused

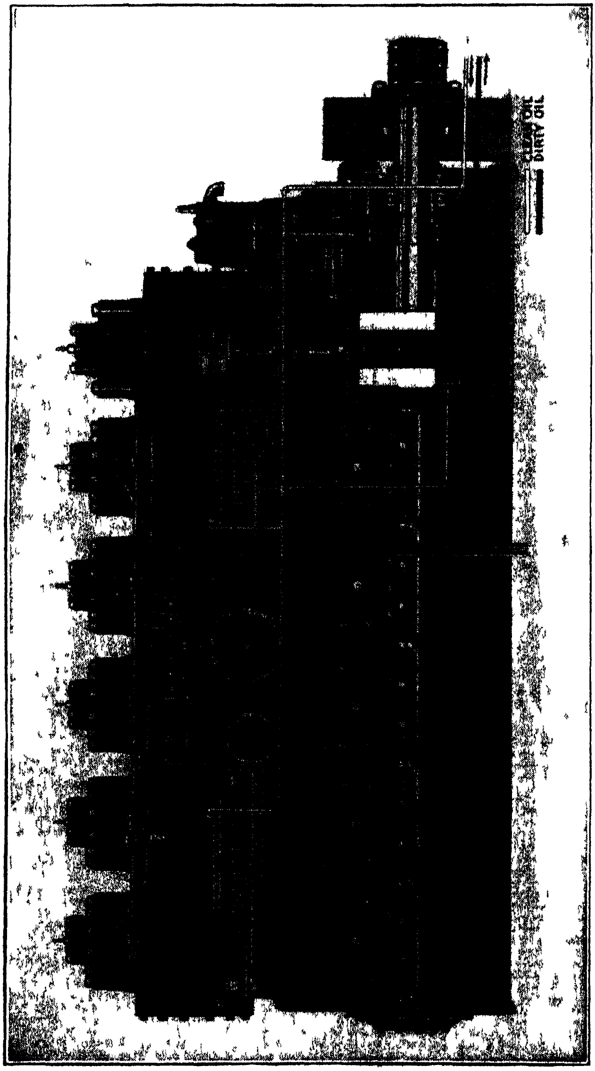


Fig. 226.—Diagrammatic Representation of Forced Feed Lubrication System, Fairbanks-Morse Marine Engine

from shaft expansion. Thus the sleeve can turn on the spherical surface and transmit power at a slight angle.

Simplification of the cooling water connections has been attained by the use of cast iron headers. There are two of these headers, one at the bottom of the cylinder water jackets and the other connected to the cylinder heads. In temperate climates the lower header valves may be throttled down, so that the main circulation takes place in the cylinder head, but in warm climates the lower header valves are partially opened. The lower header is also used as a drain for the cylinder water jackets. From the transverse section of the engine it can be noted that the exhaust manifold is connected with the cooling system.

A large 2-way ball thrust bearing is located on the after end of the crankshaft. This bearing is lubricated under pressure, and an oil throw ring and drain keeps any lubricating oil from creeping along the shaft and being thrown about the engine room.

The camshaft is driven by a train of three solid forged steel spur gears. The drive gear is keyed to the camshaft and transmits power to the camshaft gear by means of an idler gear running on a hardened and ground steel shaft supported by a cast iron jacket, which in turn is bolted and doweled rigidly to the upper base. Each meshing point of the gear train is lubricated by an oil spray.

The Bethlehem Large Unit 2-Cycle Diesel Engines. The Bethlehem large unit 2-cycle engines are of the vertical, single acting, direct reversing, air injection, cross-head type, built in sizes ranging from 1,600 to 5,000 B.H.P. and in units of 4, 6 and 8 cylinders suitable for marine and stationary service.

The basic idea in designing the Bethlehem Diesel unit was to create an engine in which all parts subject to flame temperature would be able to resist successfully the combination of temperature and pressure stresses inseparable from the operation of oil engines under heavy load. This has been accomplished by distinctive features of the design of the power cylinder, illustrated in the sectional view of the engine, Fig. 227 and in the photograph of the cylinder, Fig. 228. It will be seen that the inner and outer walls of the cylinder are united only at the top and at a point removed from the heat of combustion. By means of the flanges on the outer cylinder

walls, the cylinders are mounted in pairs in a cylinder support casting, Fig. 229, designed to receive them. The power cylinder projects downwardly through the cylinder support, to which it is attached only at the flange.

As Fig. 228 shows, the cylinder is cast with the water jacket space very open and accessible. This opening in the outer jacket wall is closed by a light cast iron sleeve attached at its upper end to the power cylinder by a flange and provided at its lower end with a water joint permitting axial expansion without danger of leakage. This sleeve is shown clearly in Fig. 230. As the power cylinder, the piston, and the outer water jacket wall are all able to expand freely in a longitudinal direction, there is no trouble whatever from difference in the expansion of any of these parts. The cylinder construction also provides for the adequate inspection and cleaning of the entire water jacketed surface.

It will be noted that the exhaust ports are cast integrally with the cylinder barrel, thus avoiding any water joint between the cylinder barrel and the exhaust ports. The slightest water leak through such a joint would produce troublesome corrosion when low grades of fuel, high in sulphur, are used.

By referring to Fig. 227, it will be seen that the cylinder is held down to its cylinder support by studs passing through the two flanges of the outer cylinder walls. With the construction of the Bethlehem Engine as shown, it is evident that neither the cylinder barrel nor the exhaust ports are subject to axial tensile stresses.

In an internal combustion engine stresses tending to burst the cylinder cannot be safely reduced by increasing the thickness of the cylinder walls, as each square inch of the walls, particularly at the upper end of the cylinder, must pass a large amount of heat to the water jacket in a very short space of time. The greater the wall thickness, the greater the consequent difference between the temperatures of the inside and outside surfaces of the cylinder walls. The greater such difference in temperature, the quicker a cylinder will fail, due to heat fatigue.

The practical success of an internal combustion engine

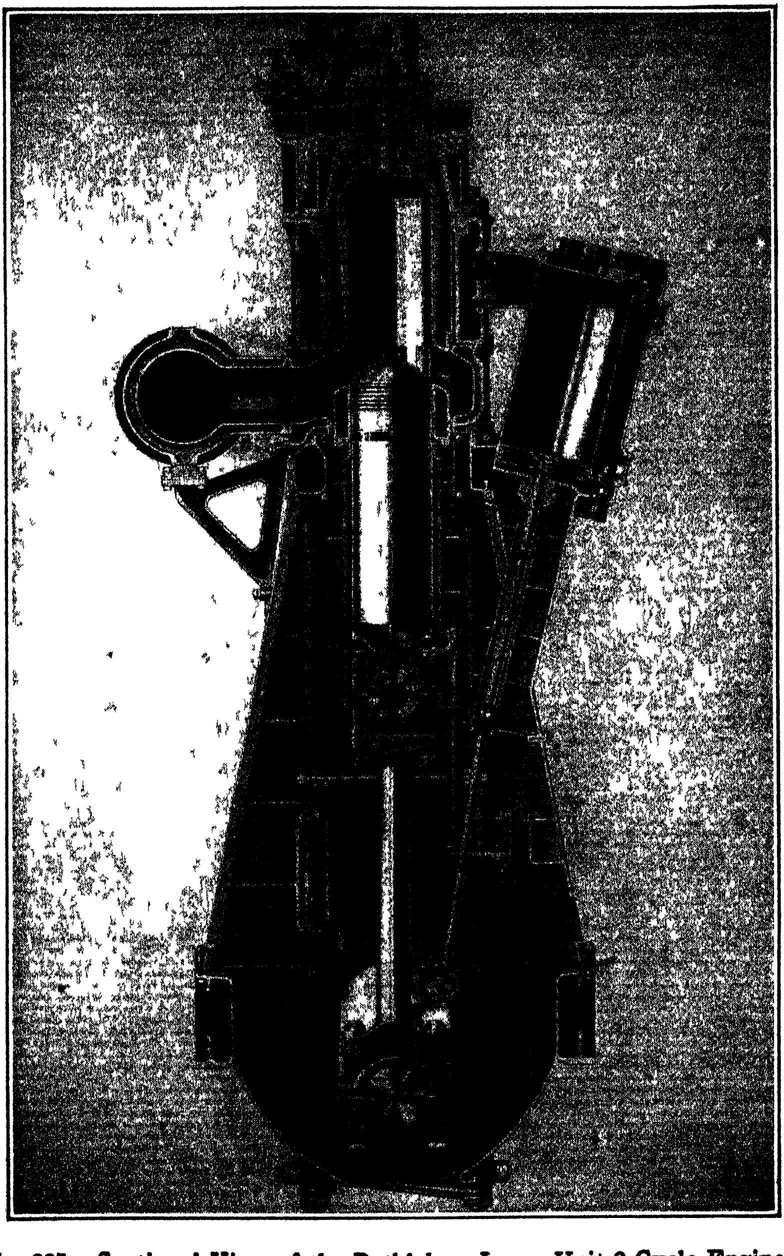


Fig. 227.—Sectional View of the Bethlehem Large Unit 2-Cycle Engine

depends upon the avoidance of such heat fatigue stresses, and the necessity of care and skill in the design of cylinders, nat-

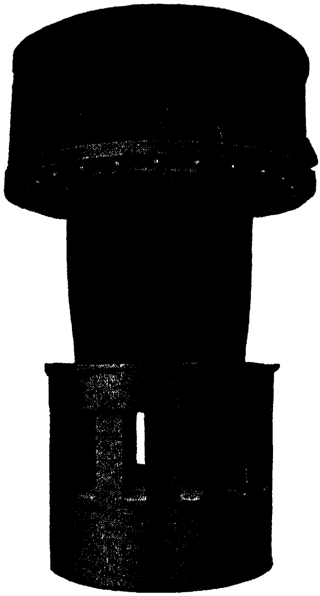


Fig. 228.—Power Cylinder Showing Exhaust Ports Cast Integral with the Cylinder Barrel. The Large Area for Water Jacket and the Mounting Flange, Bethlehem Engine

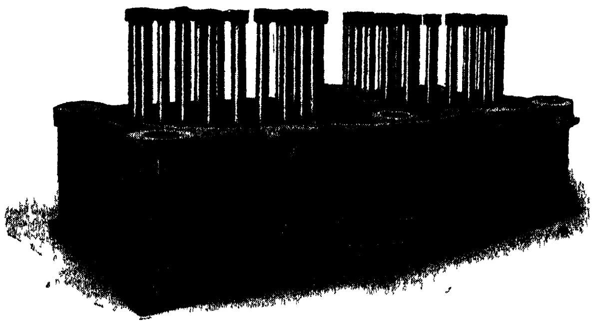


Fig. 229.—Cylinder Support for Two Power Cylinders, Bethlehem Engine

urally increases with the size, since thickness of walls must increase with the diameters of power cylinders. To cover

this point the Bethlehem power cylinder has been designed as shown in Fig. 227 and in the larger scale in Fig. 230. The upper part of the cylinder is reduced in size from the diameter of the bore to that of the scavenging valve cage. This form, as compared with an open ended cylinder, increases the strength without necessitating an increase in thickness of the walls.

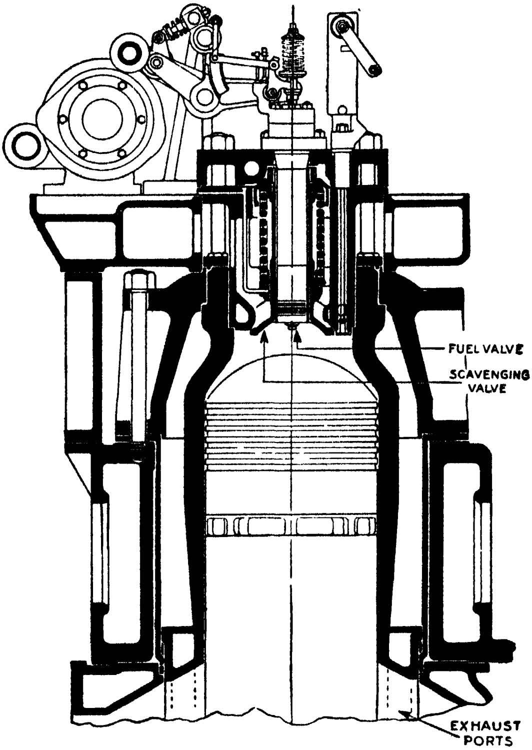


Fig. 230.—Section Through Valve Cage, Cylinder Head and Power Cylinder, Bethlehem Engine

It will be noted from Figs. 227 and 230 that the cylinder wall thickness is uniform in all that portion of the cylinder subject to maximum temperature and pressure, the thickness decreasing with the lower pressures in the lower part of the cylinder. Such uniformity in section causes uniform change in shape when heated.

The part of the cylinder subject to greatest heat is that portion between the lower end of the scavenging valve cage and the portion of the wall covered by the piston rings when the piston is at or near top center. By reference to Fig. 230 it will be noted that this portion is efficiently cooled by the water jacket, which extends around the cylinder wall above the level of the bottom of the scavenging valve cage. The hot cylinder walls thus efficiently cooled by water, are also cooled on the inside surface by the scavenging air entering through the mushroom shaped scavenging valve. The shape of the cylinder is such that the incoming current of scavenging air passes directly along this heated wall surface, absorbs from it a large amount of heat, and assists in cooling the hottest portion of the cylinder walls. Thus the cylinder is freed from the danger of heat fatigue by three means:

1.—Cylinder walls are of very strong form, with moderate and uniform thickness in the upper portion.

2.—They are efficiently cooled on the outside surface by circulating water.

3.—The walls are air cooled on the inside surface by the scavenging air. Cooling done by this scavenging air not only helps to maintain the cylinder wall at a safe temperature, but it also increases the efficiency of combustion, since heat absorbed by the air from the cylinder wall is added to the working cycle of combustion.

The only other parts exposed to maximum temperature and pressure are the power piston, scavenging valve cage, scavenging valve and fuel valve. Of these, the power piston, scavenging valve cage and fuel valve seat are water cooled, while the scavenging valve is positively cooled by the incoming scavenging air.

All parts subject to heat, viz., cylinder, piston, scavenging valve and fuel valve, are all exactly symmetrical with reference to the axis of the cylinder and each has a uniform wall thickness and is efficiently cooled. Under the varying temperature conditions existing in the power cylinder these parts expand and contract uniformly and the heat is disposed of in a manner which entirely prevents heat fatigue of any part.

The fuel valve, scavenging valve, relief valve, and air

starting check valves are contained in a single compact and easily removable cage, shown in Figs. 231 and 232.

Complete combustion in any oil engine requires that the power cylinders be efficiently cleared of exhaust gases. The clearance space in a four-cycle engine cylinder cannot be cleared of exhaust gases. Due to the fact that at the end of the exhaust stroke these gases must be at a pressure far enough above atmospheric pressure to cause them to flow from the exhaust pipe, it follows that even in the most carefully designed four-cycle engine a material percentage of the suction volume of the cylinder is occupied by exhaust gases, the presence of which is necessarily detrimental to combustion. In the Bethlehem design of two-cycle engine the scavenging air enters through a central scavenging valve in the extreme upper end of the cylinder, and sweeps downward, filling the power cylinder and driving out the burnt gases, which escape through exhaust ports located around the entire circumference of the lower end of the power cylinder. It will be evident that this positive operation must clear the cylinder of all burnt gases, and is superior to any method by which scavenging air is both admitted into and exhausted from the bottom of the cylinder.

When using low grade fuels, a certain amount of residue is sometimes deposited on top of the pistons. With the scavenging arrangement described such residue is blown out into the exhaust passages before it has time to be hardened by heat. It should also be noted that with this system perfect scavenging can be obtained, no matter how long the stroke may be for a given cylinder bore.

The small bore, long stroke engine is more economical in fuel and lower in repair cost than an engine in which the stroke and bore are more nearly alike. The gain in economy and repair costs by building an engine of a given horse-power of small bore, long stroke, and low revolutions has always been evident in steam reciprocating engines, but is still more important in internal combustion engines, due to the higher temperatures and pressures necessarily employed.

For oil engines employed in marine service there exists also the advantage of slow revolutions, as in this way a better

propeller efficiency can be obtained. For merchant ships single screw installations are cheapest and most efficient. It

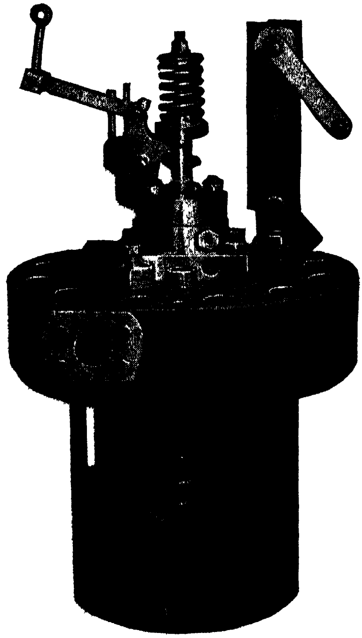


Fig. 231——Valve Cage, Bethlehem Engine

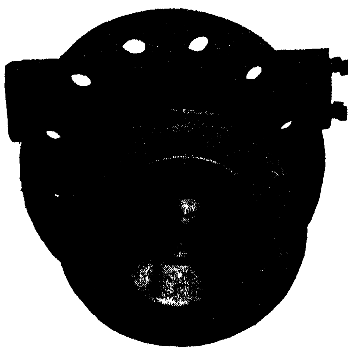


Fig. 232.—Bottom of Valve Cage, Bethlehem Engine

is worthy of note that the Bethlehem long stroke two-cycle engine is especially suited for ship propulsion because the

efficiency of both engine and propeller are highest at low revolutions.

The scavenging compressors are arranged on an incline, and are driven from the main connecting rods, as shown in Fig. 227. This inclined scavenging compressor drive has been proved by long, continued, hard service to be an unqualified success. Its advantages are several. It does not lengthen the main engine, and still more important, especially on a ship, it does not increase the number of crankshaft bearings which must be kept in line. Furthermore, this arrangement of scavenging compressors does not take up any floor space, does not increase the height of the engine, and does not materially increase its weight. These points are all important aboard ship.

By casting the piston of aluminum alloy and by using a hollow piston rod, the weight of the reciprocating parts is reduced to a minimum. Due to its lightness the scavenging piston bears very lightly on the lower side of the cylinder wall. As the total forces due to one scavenging compressor are less than 2 per cent. of those on the main connecting rod from which the compressor is driven, it is evident that no more wear will be found on this main crank pin than on the others.

The general design of the engine is based upon the unit system, each unit consisting of two power cylinders, cylinder support, A-frames, bed-plate section and crankshaft section. By combining these units the engine can be built as a four, six or eight-cylinder motor. The crankshaft sections are interchangeable, each consisting of two cranks, set 180 degrees apart, with flanges on each end. The timing of cranks needed for different combinations of cylinders is obtained by bolting the shaft sections together in the required positions.

The construction of the crankshaft sections with the cranks opposite has two advantages:

First, the inertia forces of the reciprocating parts attached to these cranks are, except for the angularity of the connecting rod, equal and opposite. The vertical planes in which these opposing and counterbalancing forces act are

closer together than is possible with any other crank arrangement. The comparative closeness of these planes of action therefore keeps down to a minimum the tendency of an engine to produce periodic vibrations in a ship.

Second, the advantage in constructing a shaft section with two opposite cranks is very important from a practical point of view. The whole crankshaft section is in one plane so that a center section of crankshaft can be removed endwise from a six or eight-cylinder engine without disturbing either the other crankshaft sections or any part of the A-frames. The injection air compressor is driven from the forward end of the engine. To preserve the interchangeability of crankshaft sections, the small crank driving the injection air compressor is bolted to the forward flange of the forward section.

An important feature of the construction of the Bethlehem Oil Engine is the design whereby the major load-bearing parts, such as the cylinder supports, A-frames and bed-plates, are subjected to compression loads only instead of both tension and compression loads. This insures construction that is light, strong and rigid.

Each cylinder support contains a pair of power cylinders and rests on A-frames, to which it is rigidly held by long bolts passing from the top of the cylinder support through the A-frames and bed plates. This construction is shown in Figs. 227 and 233. When the engine is erected, these long bolts are tightened to produce in them an initial tension about 25 per cent. greater than the maximum tension arising from combustion pressure in the cylinder. The bolts are of such a size that this initial tension is not more than 20 per cent. of the elastic limit of the material. As the initial tension in the bolts is more than the maximum tension due to combustion pressure, it follows that the bolts do not stretch in the slightest degree while the engine is working, and the A-frames are therefore always in compression. Since the cylinders are held to the bed-plates only by these large bolts passing through the A-frames, the latter are never subjected to tension. This construction gives the utmost rigidity without undue weight, as the cast iron A-frames need be designed only for compression, the large bolts absorbing all the working stresses.

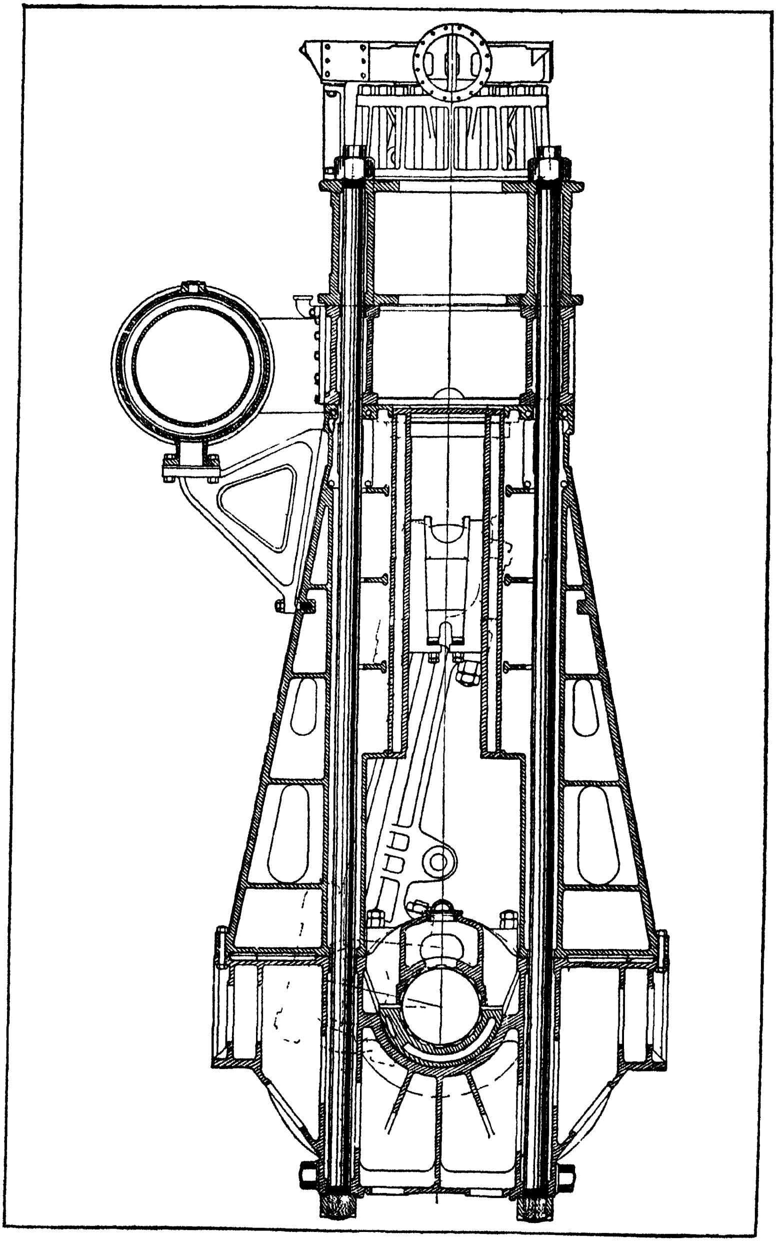


Fig. 233.—Framing of Bethlehem Engine

By referring to Fig. 233, it will be seen that in the Bethlehem Oil Engine the large bolts pass clear through the bed-plate quite close to the main bearing, the distance from center to center of bolts being less than half the width of the bed-plate. The part of the bed-plate subject to stresses arising from combustion pressures is only the portion between these bolts, and, therefore, the span of bed-plate subject to such stresses is less than one-half, the stresses in the bed-plate are less than one-quarter and the deflection of the bed-plate is less than one-eighth of those of ordinary design.

The combustion pressures on the power piston, acting on the bed-plate, set up tension stresses in the lower side of the bed-plate between the large bolts. To resist these tension stresses, the bed-plate is provided in its lower part with the horizontal bolts shown in Fig. 227 and 233. During erection, these bolts are subjected to an initial tension about 25 per cent. more than the maximum stress due to the combustion pressure. Therefore, no deflection of the bed-plate can occur and there is no possibility of a cracked bed-plate due to shrinkage strains. No part of either the A-frames or bed-plates is subject to tension due to working stresses, such stresses being entirely taken up by forged steel bolts.

In the usual oil engine construction, the power cylinders are bolted to the A-frames, the feet of which are bolted to the bed-plates near the outer edge. This arrangment subjects the whole width of the bed-plate to bending stresses arising from combustion pressures, with consequent deflection of the bed-plate.

Compressed air for injecting oil into the power cylinders is supplied by a three-stage, vertical air compressor which is securely bolted to the forward end of the engine frame and driven from an overhung crank bolted to the forward end of the crankshaft.

This compressor is shown in section in Fig. 234, and is of the differential piston type, having a double-acting first stage and single-acting second and third stages. The valves are of the automatic disc type, which are efficient and durably constructed with ample areas, readily accessible and easily

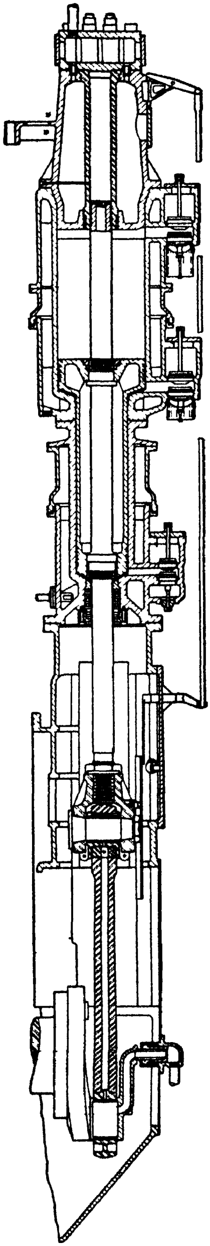


Fig. 234.—Section Through Air Compressor, Bethlehem Engine

removed. The air is cooled by an efficient cooler after each of the three stages.

The engine controls and gage board are shown in Fig. 235; and the fuel pump is shown in Fig. 236.

The left hand lever controls the master valve which supplies starting air to the engine. Properly timed valves admit this air into the individual cylinders, causing the engine to start. Fuel is then injected into the cylinders and when combustion takes place, the starting air is automatically cut off by check valves in the cylinders, and the master valve may be closed. Rapid and positive starting is obtained with this system, by which fuel oil and starting air may be used on all cylinders at the same time.

The middle lever controls the amount of fuel injected into the cylinders and works in a double slot somewhat similar to the selective gear lever on an automobile. The amount of fuel is increased by pushing the lever away from the operator. When the operator wishes to reverse the engine he pulls the fuel lever towards him as far as he can, shutting off fuel and stopping the engine. By moving the lever into the astern slot, compressed air is admitted to the cam shaft reversing mechanism, which sets all cams in the proper position for the reversal of the engine. Advancing the lever further into the astern slot sets the throttle for the required amount of fuel. At the same time the engine may be started with compressed air by moving the extreme left hand lever upward. As soon as fuel in the cylinders is ignited, the starting air lever is returned to its original position, shutting off the starting air, leaving the engine entirely controlled by the single middle lever, or throttle.

It will be seen that this reversing gear is practically foolproof for the following reasons: The engine cannot be reversed without first entirely shutting off the fuel; the engine cannot be given fuel in the reverse direction until the reverse gear has completely moved into the proper position.

The right hand lever determines the lift of the fuel valve so as to properly control the injection air at all speeds and powers, as previously explained. This arrangement has the additional advantage of allowing the engine to be started, in

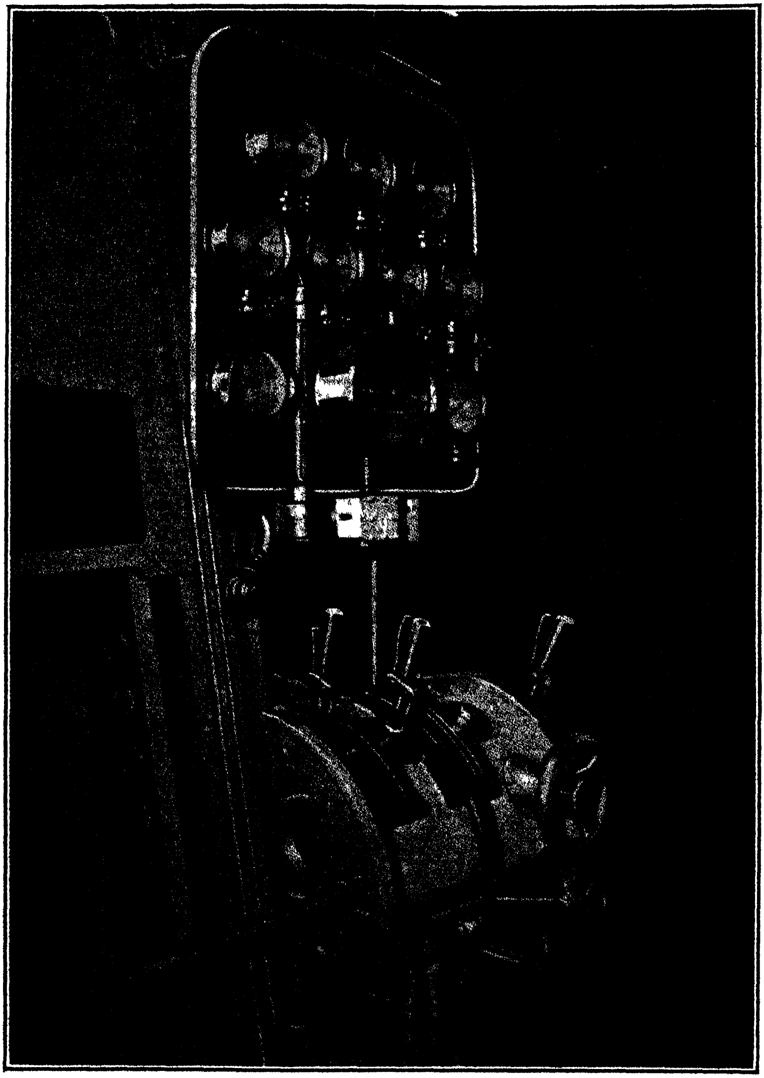


Fig. 235.—Engine Controls and Gauge Board, Bethlehem Engine

emergency, without the necessity of charging the injection air bottles beforehand, as injection air may be built up very rapidly by the engine air compressor.

The Bethlehem Small Unit 2-Cycle Diesel Engine. The Bethlehem Oil engine, small unit type Fig. 237, is of 2-cycle, vertical, single-acting, heavy duty, solid injection type working on the full Diesel Cycle with cylinders efficiently scavenged by air supplied by a double-acting scavenging pump driven off the main crankshaft. The engine is of the fully enclosed type, fitted with force feed lubrication and is suitable for either marine or stationary work. It is built in units of 3, 4, and 6 cylinders and in sizes ranging from 60 to 570 B.H.P.

The engine bed plate and housing are made from the best grade of cast iron and well arranged to carry the working stresses to which they are subjected. The housing or crank case of the larger unit is bolted to the bed-plate, but the main working stresses for each of the cylinders are carried by four forged steel tie rods running through the housing frames to the main bearing arches. These tie rods therefore take the working pressures and are keyed at the bottom to prevent turning when setting up the nuts. The upper half of the housing is arranged to carry the cylinders and to also form a large receiver for the scavenging air.

The pistons are of the latest design and made of a grade of cast iron specially developed for this service. The piston crown is of special section which insures the conducting of sufficient heat to the cylinder walls to permit the piston to operate at a safe temperature without artificial cooling. A suitable number of cast iron snap rings are used at the top and one scraper and seal ring is used at the bottom of each piston.

The cylinders and cylinder heads are properly designed for water cooling with the jackets cast integral with each part. The engine is arranged for port scavenging and exhaust and scavenging ports of ample area are located near the bottom of each cylinder. All water joints that are apt to allow water to leak into the cylinders or crank case have been eliminated.

The crankshaft is of the solid forged steel type made of high grade open hearth steel and is interchangeable with that of any other unit of the same size.

The connecting rods are of open hearth forged steel. They are designed with an eye at the top and a tee foot at the bottom end to which is bolted the cast steel crank pin boxes lined with a high grade of babbit.

Among the features of the Bethlehem small unit type engine that assure simplicity and reliability in operation are the following:

The Bethlehem engine is simple in construction and oper-

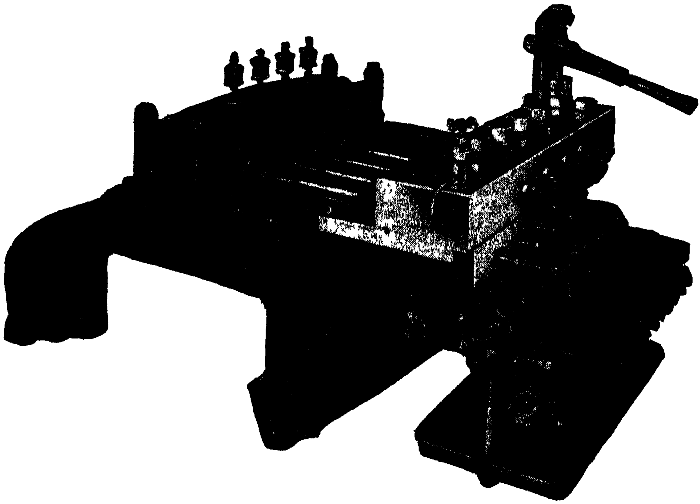


Fig. 236.—Fuel Pump of Bethlehem Engine

ation. Moving parts have been reduced to a minimum, the cylinders having no mechanically operated valves or camshafts. Starting is especially simple. A wide range of fuel oils may be burned readily and the rate of fuel consumption is low.

The Leissner system of fuel injection is used. This system combines the advantages of airless or solid injection together with those of the constant pressure cycle and eliminates mechanically operated spray valves and injection air compressors.

Fuel timed and controlled by the fuel pump is injected directly through the automatic spray valve into a duplex combustion chamber of special design. Ideal combustion conditions

permit heavy as well as light grades of fuel oil to be burned efficiently and at low consumption rates.

The engine is started by compressed air, through a simple control device which allows both starting air and fuel oil to be applied to the cylinders at the same time. This method eliminates the complicated mechanism necessary for applying air, separately to some cylinders while others are getting oil.

An attached double-acting scavenging pump furnishes an excess amount of scavenging air for thoroughly cleaning the burned gases from the cylinders. This type of pump is very satisfactory for efficient and reliable operation. It is readily accessible, easily maintained and requires practically no attention.

The burned gases are exhausted and scavenged through ports in the lower part of the cylinder resulting in the entire elimination of valves, camshafts and rocker arms for this purpose.

The cylinders, cylinder heads and exhaust header are cooled by water from the cooling pump to maintain uniform temperatures.

Forced feed lubrication is used for all the bearings and cylinders of the unit. The smallest bearings are supplied with a stream of oil from the main lubricating oil pump assuring an adequate and dependable supply of oil at all times.

Nelseco Solid Injection Engines. These engines are built in two cylinder sizes with diameters of 12 inches and 14½ inches, with a stroke of 18 inches and 22 inches respectively. The power output ranges from 100 in the smaller size to 500 B.H.P. in the larger. Engines are constructed in convenient units of two, three, four and six cylinders. The engines use the M.A.N. system of timed injection and are built both reversible and non-reversible. A three cylinder unit is shown in Fig. 238 and sectional views in Figs. 239 and 240.

Engines of this design have many good points. Some of these, which are especially well taken care of are rigidity, accessibility, and interchangeability of parts. Starting is accomplished with very little effort as mechanical injection engines may be started easily by admitting fuel at the same time as the starting air, and also because it is unnecessary to turn the

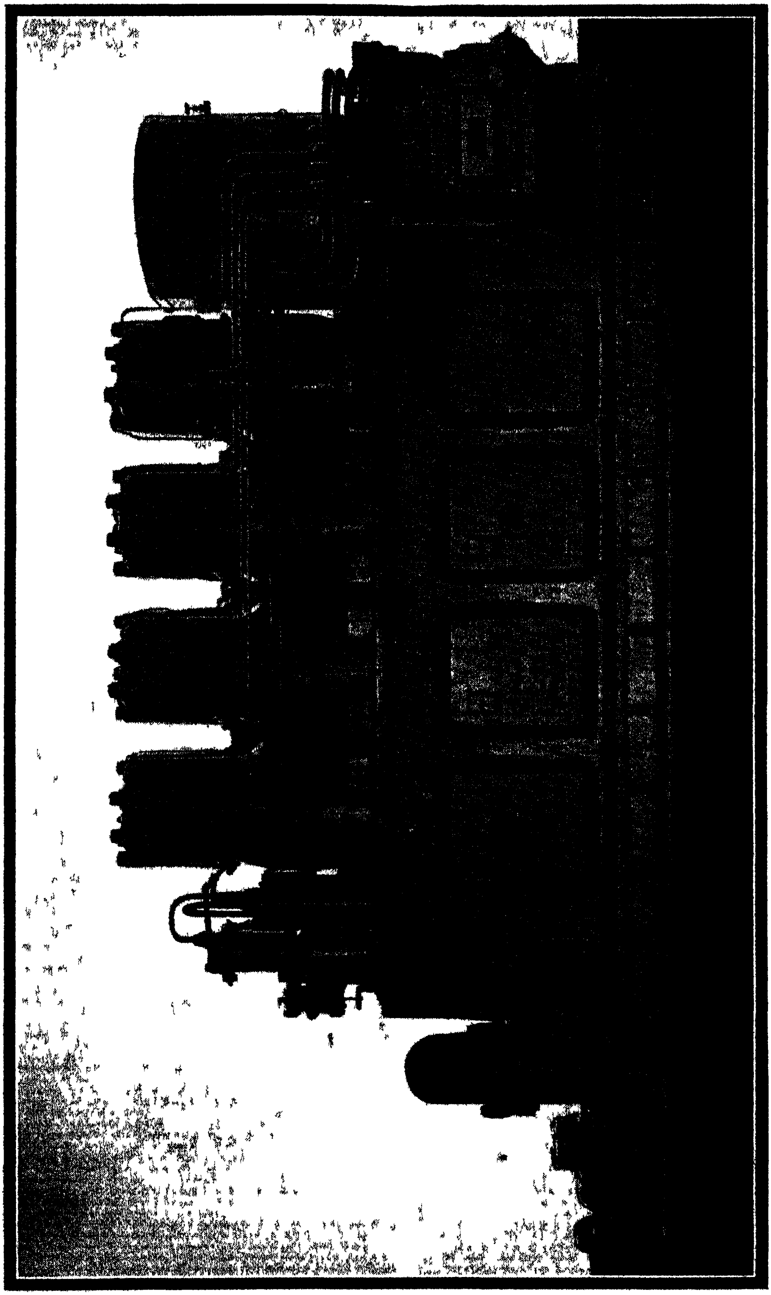


Fig. 237.—Bethlehem Small Unit Type, 2-Cycle, Solid Injection Engine

engine over a sufficient period to build up an injection air pressure. Extremely low fuel consumptions have been obtained,

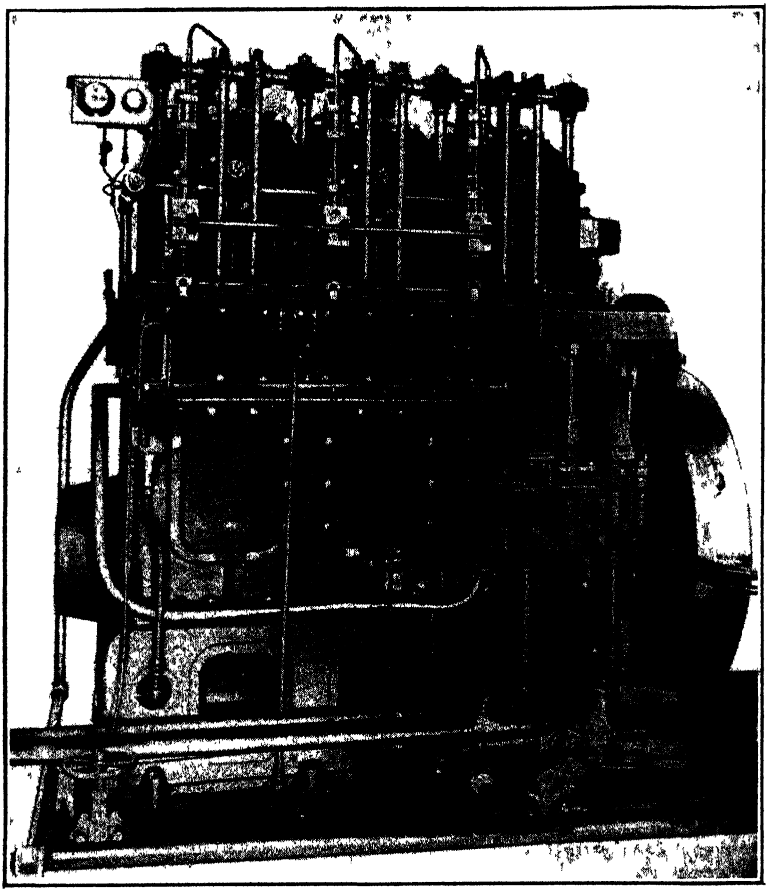


Fig. 238.—Operating Side of Nelseco Solid Injection Engine

a figure of .37 lb. per brake horsepower per hour having been obtained in recent tests.

The engines as a whole are of pleasing appearance, which together with smooth running, economical fuel consumption, and ease of maneuvering, renders them as exceedingly useful units.

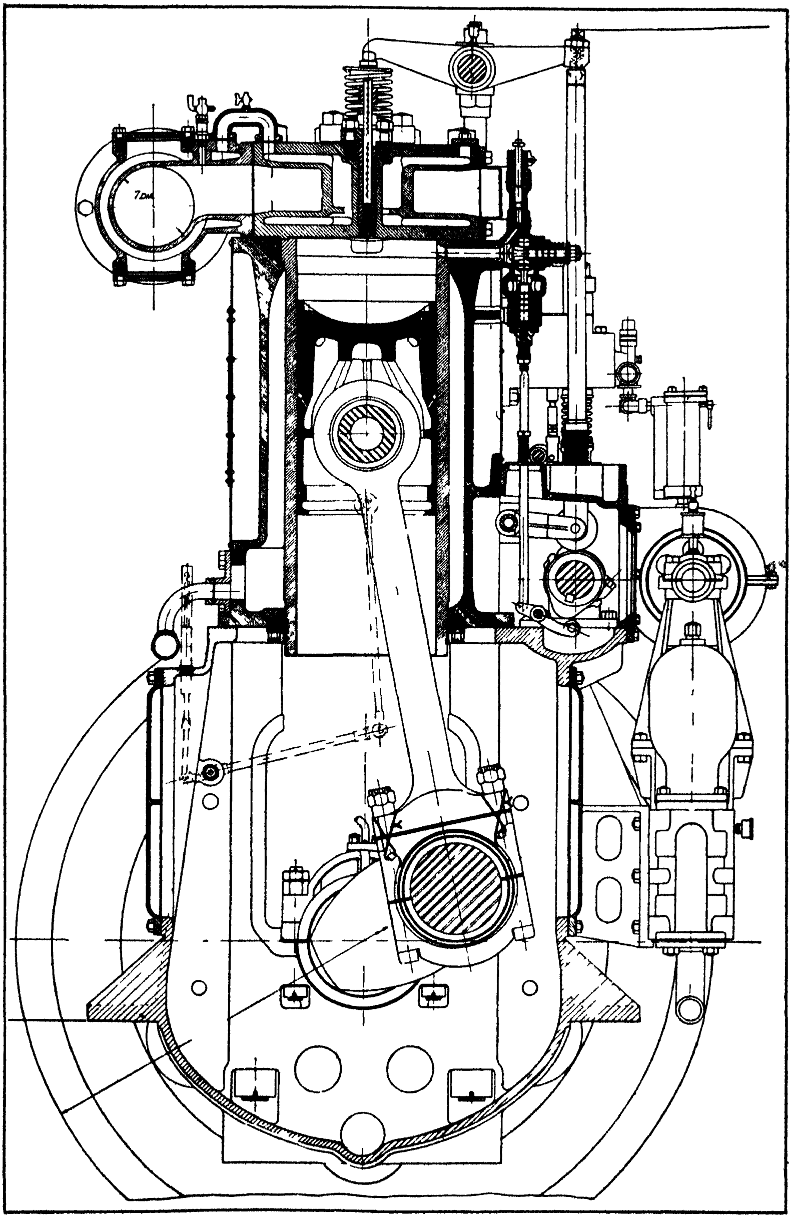


Fig. 239.—Cross-Section Through Nelseco Solid Injection Engine

All sizes follow the same general design as far as the appearance of engine is concerned. In the 12″ diameter size, the bedplate and housing or columns is made in one casting, on which the cylinder block is mounted, whereas in the large

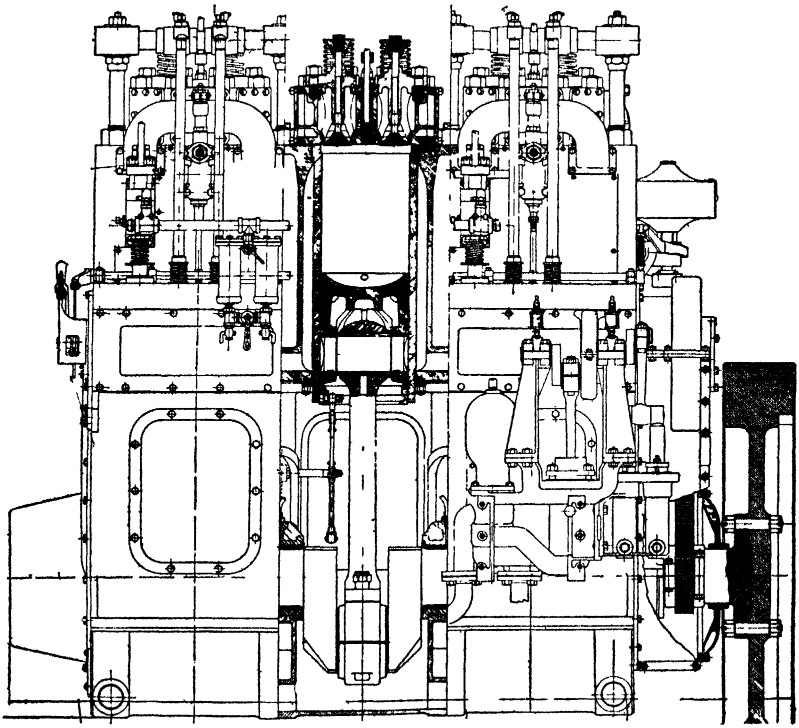


Fig. 240.—Front Elevation of Nelseco Solid Injection Engine

size of 15″ diameter a deviation from this practice is made and a separately cast housing is interposed between the bedplate and cylinder block.

The cylinder block, Fig. 241, consists of one rigid casting into which the liners are inserted. These liners are plain cylindrical castings and are arranged in the cylinder block with a gland at the bottom to prevent any leakage of jacket water. The upper end of the liner is held securely in place, while the lower end in passing through the gland is allowed to move freely. This construction allows for any expansion that may

take place in a vertical direction. Starting air is introduced in the cylinder through an opening drilled above the water jacket lip in the liner, and below the cylinder head joint. In this manner the number of openings in the cylinder head is reduced, all of which tends toward a more uniform head casting (see Fig. 239). The space on the cylinder block which is not occupied by the liner or acting as a water jacket is enclosed

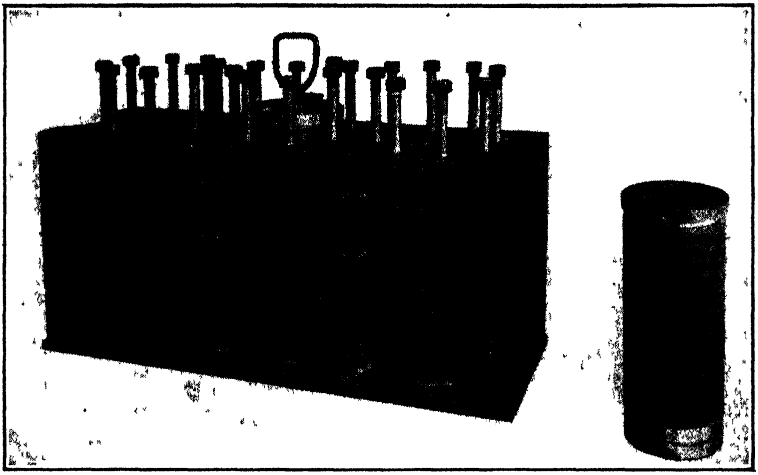


Fig. 241.—Cylinders of Nelseco Solid Injection Engine Showing Liner at Side

by a sheet steel casing. This casing is fitted with screened covers, and acts as a very efficient intake muffler for induction air, being connected by small cast iron ducts to the intake valves.

The cylinder head is a rugged hexagonal casting with a horizontal plate interposed between the top and bottom, dividing it into two parts; the cooling water enters at four points between the plate or disc and adjacent to the combustion space and horizontal dividing plate referred to. At this point the velocity is high and good cooling is thus obtained. The cooling water then passes upward through the dividing plate at a point in the vicinity of the spraying device and the velocity is reduced, the water finally passing through into the jacket

of the exhaust header and thence out through a single opening. The valves in the head are in separate cages and are interchangeable.

Tie rods extend up from the bedplate to the top of the cylinder block and are fitted in way of each main bearing. These rods can be seen in Fig. 242 and are utilized to take the work-

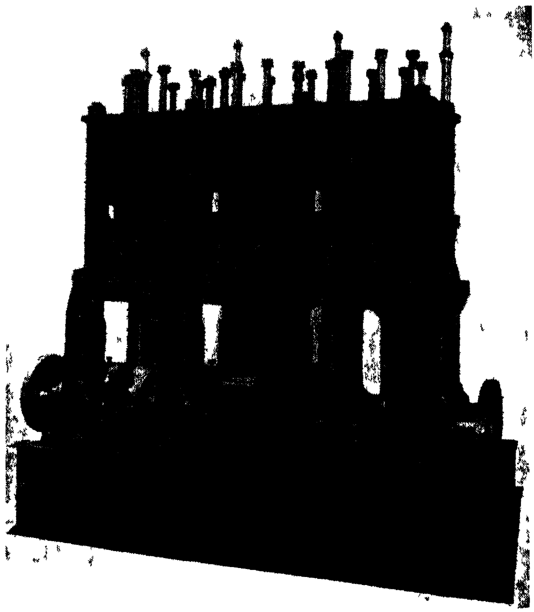


Fig. 242.—Bedplate and Cylinder Block of Three Cylinder Nelseco Solid Injection Engine

ing stresses, and are so arranged that all stresses are practically removed from the intervening cast iron structures of the housing and cylinder block. On the inboard or operating side, these rods extend upward above the block and support the shaft upon which are mounted the valve rocker arm levers. These levers act directly on the inlet and exhaust valves and are operated by vertical push rods actuated by cams through links and rollers at their lower ends. The rocker arm shaft is divided into parts corresponding with the number of cylinders so that removal is easy, and no adjustment is altered.

The cam shaft extends along the operating side at the level of the bottom of the cylinder block, and is spur-wheel driven. A spur wheel on the camshaft drives a reciprocating pump unit that is mounted on a crank case door. Fuel, lubricating oil and circulating water are thus supplied. There are three cams per cylinder, one each for the fuel pump, exhaust and inlet valve, while the latter also actuates the air starting valve. In the reversible engines, the camshaft is moved longitudinally; sloping sides of the cams being used to eliminate the necessity of removing the cam rollers from the cams when maneuvering.

The fuel pump consists essentially of a steel block with drilled passages. A pump plunger, cam-operated and spring-returned, works in a removable lapped fit bushing, eliminating the use of packing. One suction valve and two discharge valves of the usual type are fitted. The injection is controlled by lifting a bypass valve on the discharge side of pump, at the desired time, so that the starting time of injection is fixed relative to the crankshaft, but the time of stopping injection is variable, depending on the load. The bypass valve works in a lapped fit bushing which is also easily removed or replaced. A filter, in duplicate, on the suction side of the fuel pumps prevents any foreign matter over .004″ dimension from passing. From the fuel pump the fuel is led through a brass pipe to the atomizing nozzle. The nozzle consists of a metal plate with very fine holes of varying diameter and number, depending on the power to be developed and the speed of revolution. Clogging of the holes very seldom occurs because of the efficient filters, and also because of the fact that if one hole clogs or tends to clog, the pressure automatically rises due to the restriction of discharge area, and tends to clear the nozzle itself.

For taking cards, an indicator gear is fitted. This gear is mechanically driven from each working piston and is extended through the engine housing where it is easily accessible for the purpose intended.

Regulation of engine speed is controlled automatically by a Massey governor.

The Foos Diesel Engine is a product of the Foos Gas

Engine Company, Springfield, Ohio, which has been actively engaged in the exclusive design and manufacture of internal combustion engines for the past forty years.

The Foos Diesel is an American product in its entirety and embodies many novel features never before incorporated in other designs and makes and is of the four cycle solid injection type.

Fig. 243 shows a view of a 4-cylinder Foos Diesel engine and also portrays the box design of the entire frame suggesting strength and ruggedness of the unit.

The bases are massive and rigid and so ribbed with distribution of metal to provide a perfect foundation for all parts.

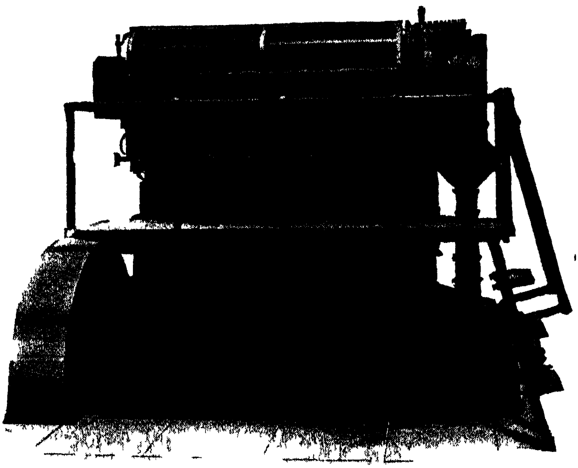


Fig. 243.—Four Cylinder Foos Diesel Engine

The bases are fully enclosed, protecting all bearings, etc., from dust and other foreign matter, also making an oil tight enclosure.

The lines of the upper base are such as to transmit the operating stresses as directly as possible from the cylinders to the main crankshaft bearings and still allow ample room and working space. The rigidity of the engine is assisted by the base design bringing the crankshaft with its working sources very close to the floor line.

The crankshaft is forged from a single billet and contains no welds or built up sections. The steel used is from open hearth close grained and heat treated and with perfect machining in its entirety.

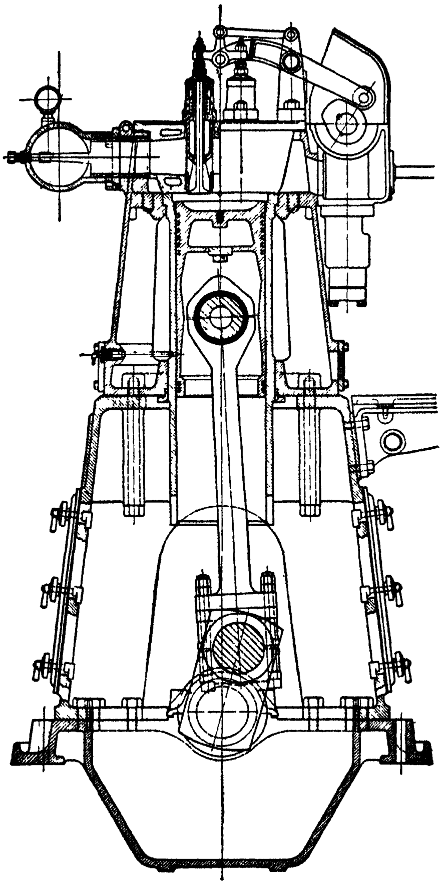


Fig. 244.—Cross-Section of Foos Diesel Engine

Figure 244 shows the form of combustion space which is essentially that of the Diesel engine, the spray valve being mounted on the cylinder axis in the same relation to the combustion chamber as a normal Diesel spray. These spray valves are comparatively simple in design, wholly automatic in

action, thus eliminating the delicate cam and lever action otherwise necessary and with one very light weight moving part. The spray valves are located in the exact center of the head and combustion space, tending to effect an exact and uniform distribution of fuel. The spray valve body is formed from a solid block of steel and all internal parts are ground to

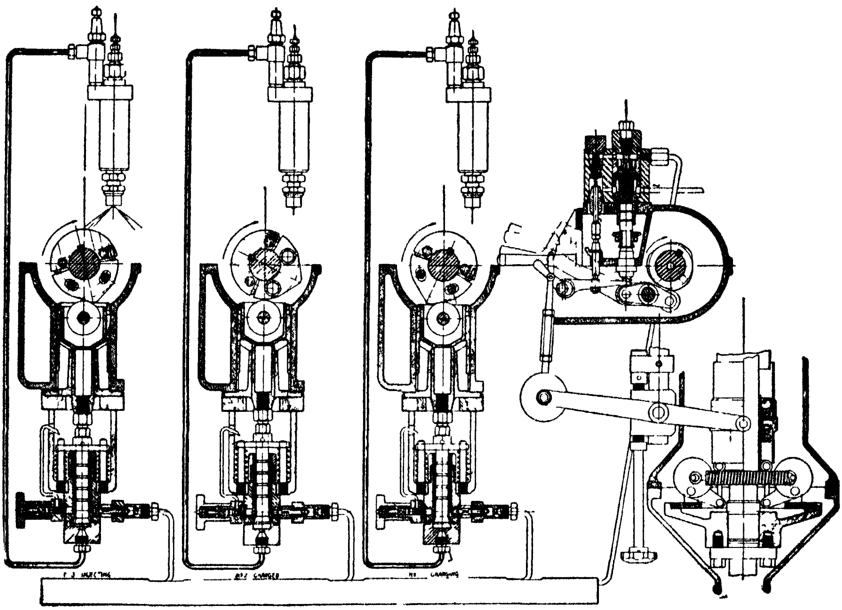


Fig. 245.—Foos Fuel Pump and Governor

a most accurate fit. The spray valves may be removed without disturbing any other part and in doing this it does not effect the timing or any other element of injection to any degree. Fig. 245 shows the fuel pump and injection mechanism in the three states of operation.

The governor is carried by the intermediate shaft, thus eliminating special gearing or driving mechanism otherwise necessary, and is so fitted with flanged couplings as to permit dismantling it without removal of shaft, gear, etc., from the base. Fig. 245 shows the cross-section of the governor while in Fig. 243 is shown the outside general appearance.

The governor is flexibly mounted, of the centrifugal flyball type, completely enclosed in an oil and dust proof case, this case being split and easily removable, disclosing the entire governor and mechanism.

The regulation of the governor is effected through the movement of a small wedge only, thus completely avoiding the transmission of even the slightest strain to any part of the governor mechanism.

The governor is designed to operate alternating current generators in parallel, as between any sizes of the Foos Diesel engines and the regulation and cyclic variation can be made such as to fit the most exacting power requirements.

The crankshaft bearings are very close to the floor and can be seen in Fig. 244 and are in the form of symmetrical shells, firmly supported by the base which is bored and hand-scraped to their exact form.

The bearings are imbedded solidly in the lower base without vertical adjustment and thus maintain an absolutely rigid alignment, under all normal conditions of operation. The shells are of bronze, machined, trimmed and lined with genuine babbitt metal, secured by dovetailed grooves, bored to size and hand scraped, to fit its portion of the shaft.

Laminated metal shims are used which permit of easy access and uniform adjustment for wear. The lower shell is interchangeable if desired, with the upper shell, the bearings are especially long and very liberal in size.

The cylinders are interchangeable and are very simple in design, with an even distribution of metal. The water jacket capacity is exceedingly large and provides for a large supply of water around the liner with comparatively high velocity at the top, in accordance with the best practice in this line. Each has four openings for inspection and cleaning. The liners in the cylinders are removable and are machined all over from high quality metal, thus avoiding a possible distortion. The liner is secured only at the top and is allowed therefore to expand as the temperature necessitates.

The packing at the lower end of the liner is adjustable or renewable without interference from the liner, cylinder or other parts and it is absolutely superior to the rubber gaskets.

The cylinders are secured to the engine base by heavy bolts located as nearly in line of working stress as possible.

The cylinders carry both inlet and exhaust manifold, camshaft, piping systems, etc., none of these being attached to the heads.

The cylinder heads are of the patented Foos type and are interchangeable and removable without disturbing either the inlet or exhaust manifolds or those of the circulating water, air starting, or other parts. The head carries duplicate inlet and exhaust valves permitting all levers and driving mechanism to be easily handled from the platform, besides large direct ports.

This design and location of valves gives an exactly symmetrical space with the minimum possible surface area and also smoothly machined, thus standardizing compression.

The use of the dual valve system permits each valve to be much smaller than the single valve and working therefore at correspondingly lower temperatures with less frequent attention or grinding and much longer life of both valve and seats.

Again with duplicate valves the lift for full area is much less than for single valves of the same area, with the wear and stresses on these considerably reduced.

The pistons, of the trunk type, are cast from a special high quality mixture and are extra long to give maximum bearing surfaces and which makes cross head construction wholly unnecessary.

Each is accurately and closely fitted to the cylinder and equipped with six pressure rings at the top and two at the lower end, so shaped as to facilitate proper lubrication.

The upper end of the piston is plain, having no special or irregular forms, deflectors, ribs, etc., which with the head entirely smooth and flat, not only improves the whole combustion process, but the distribution of metal being uniform, there is no tendency to hot spots and unequal expansion detrimental to both piston and cylinder in lubrication and inciting checks and cracking.

The piston and pin are made as light in weight as complete safety and long life permits, thus aiding smoothness of operation and avoiding needless stresses.

The injector pump, one for each cylinder, is most accessibly located in the inlet manifold with its roller in direct contact with its actuating cam.

The injector body is machined from a solid steel block with its parts very rugged in design and closely fitted.

The plungers are long and so accurately ground to fit as to avoid the necessity of packing which might tend to give sluggish action.

Each injector is equipped with a valve by which the injector is readily cut out or in, as desired, and a relief that prevents any abnormal pressure in the fuel lines.

The fuel measuring pump is located on the manifold at the governor end and operating directly from the camshaft.

Each stroke delivers the required and uniform quantity of fuel to each injector in turn which wholly eliminates the usual extremely careful attention necessary to so adjust injectors, etc., as to insure each cylinder carrying its exact share of the load and obviates the serious consequences often resulting from this lack of balance. The by-pass valve in the pump is under direct control of the governor and thus immediately responsive to any change in load.

The camshaft is solidly supported by the inlet manifold with heavy removable and interchangeable bearings fitted with bronze liners. "Split" cams are used which allow any one to be taken off without disturbing any other part.

Each cam is located by a Woodruff key which prevents any possible slipping or misadjustment of the timing.

The inlet and exhaust cams are of the "constant acceleration" type, giving smooth, quiet valve action and positive and complete scavenging of the cylinders.

The camshaft is readily dismounted with its complete assembly, or any cam or part thereon removed independently.

All bearings relating to the camshaft are oiled from the central force feed system, thus making detailed attention to this unnecessary.

The camshaft, fuel distributing pump, cylinder force feed lubricator, injectors, etc., are all carried by a heavy rigid box type manifold bolted firmly to the front of all the cylinders

at their upper ends, thus naturally adding also to the general rigidity of the engine.

This arrangement muffles the air thoroughly and also permits taking in air at one point only—the flywheel end of the manifold, and thus readily piping the air used by the engine from outside the engine room if desired.

Cleanliness is insured by all surplus lubricating oil from the upper portion of the engine being collected by this manifold and returned to the pressure system.

The vertical intermediate shaft is driven from the crankshaft through spiral gears and in turn transmits its motion to the camshaft by the same means. One gear of each pair is steel, the other bronze and each practically runs in oil supplied by the complete pressure system.

The only one used other than these is a plain spur gear driving the lubricating oil pump and all of these are of selected material and accurately cut on machines designed for the purpose.

The lubricating oil is forced under pressure by a pump driven direct from the crankshaft, into all main crankshaft bearings, crank pin, piston pins, intermediate shaft thrust bearings, gears, etc.

All camshaft bearings, valve levers, rollers and pins, etc., are covered by the same system, leaving practically no parts on the engine requiring oiling by hand.

After passing through the bearings the oil is passed through a duplicate strainer, either side of which can be cleaned while the other is in operation and from this is pumped back into the pressure lines.

The outboard shaft bearing is fitted with a ring oiler bearing having a large reservoir with a convenient gauge indicating the oil level.

Each cylinder is fitted with two oil lines from a force feed lubricator positively driven from the camshaft, this being entirely independent from the general pressure system, and its feed adjustable while in operation.

The governor receives a constant stream of oil from the pressure system which after working through the whole gov-

ernor mechanism and bearings returns automatically to the system.

The Foos force feed system therefore takes care of virtually every bearing and wearing point on the engine using the oil over and over at high pressure and speed.

Starting air is connected to each cylinder and each so controlled by a cam as to allow air to reach the starting valves only during the working stroke. It cannot be gotten into the cylinder at any other time.

The valve design is such that starting air can enter the cylinders only when its pressure is greater than in the cylinders and which avoids the danger in other designs of building up accidental and disastrous pressure.

This automatic feature also insures utilizing the power stored in the regular compression in the cylinders as starting air does not enter until this compressed air has expended its force on the pistons.

With this system the fuel valves are open in starting and thus regular ignition and combustion may be taken up in each cylinder on any working stroke, which action shuts off the starting air from that cylinder. The action is in every detail completely automatic.

This system completely avoids the complexities and skill required by the usual design which necessitates shutting off the starting air by hand before fuel oil can be safely turned on or allowed to enter.

CHAPTER XII

A 1,000 Horse Power Submarine Diesel Engine

Main Engine Lubricating System—Piston Cooling System—Bearing Lubrication System—Cylinder Lubrication—Air Compressor—Fuel System—Fuel Injection Valves—Engine Control.

The engines used by the Germans to propel their submarines are probably the most highly developed Diesel Engines ever manufactured in quantity. About 500 of the 1,200 H.P. type were used in their 800-ton submarines, besides many of greater power for larger boats, including the 3,000 H.P. type used on the 2,700-ton submarines. These engines, as well as smaller types, are all somewhat similar as to general and detailed design; they are all 4 cycle single-acting air-injection engines. The extraordinary reliability and flexibility of these engines were a source of considerable surprise to those present at the time these submarines were surrendered. It was noted that the engines ran smoothly at but 1-10 of their maximum speed, and even while idling with no load at this low speed they had a clear smokeless exhaust. To still further emphasize the performance of these engines, a comparison with ordinary Diesel engines shows that the power developed for a given size cylinder is over twice as great for the former. This means that twice as much heat is produced in a given space with the result that the exhaust temperature is about 1,000° F. while ordinary Diesel engines have but 600° to 700°. This difference in temperature greatly increases the mechanical difficulties resulting from heat which has necessitated the use of oil cooled pistons as well as water cooled exhaust valves and exhaust valve cages.

Engines of similar design are being manufactured at the Navy Yard, New York. The Navy Department has been kind enough to give us access to drawings, photographs and test

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data of these engines, which have been made use of in preparing this article. It is so seldom possible to publish much of the minute technical details responsible for the success of such a perfected mechanism, that this data has been given for those who can make use of it.

It must be borne in mind, however, that submarine engines are so highly specialized for their particular service that they can hardly be considered as commercial types particularly on account of their higher cost per horse power. Several of these engines have nevertheless been installed in the hulls of merchant ships, and even though their r.p.m. has been reduced somewhat it has been too high for the propellers of these ships and reduction gearing has been used. The use of engines with such high power development for their size will be followed with intense interest by those who believe that reliability is not necessarily sacrificed by the high power development and speed, as well as by the comparatively light weight of these engines.

As perfection of the details of the lubricating system has had much to do with the extraordinary success of these engines, particular study is given in this article, to these details as well as to the handling and control of the fuel. It is not possible here to go into the interesting and ingenious mechanical details concerned with their construction, though much can be learned from the sectional drawings shown in Figs. 246, 247, 248.

Main Engine Lubricating System. Every moving part of the engine except the valve stems and part of the maneuvering gear, is lubricated from a full pressure circulating system which reaches even the rollers on the valve rockers. The completeness of this circulating system reminds one of the lubricating systems developed especially for airplane engines, and as will be seen later, the bearing clearances and grooving are also similar to what is considered as best airplane and racing engine practice. The few parts not lubricated by the circulating system are supplied with oil and grease cups, which are sufficient since the maneuvering gear does not require much oil. Even if the lubrication of these parts is neglected there will be no other ill effect than making the work of the operator

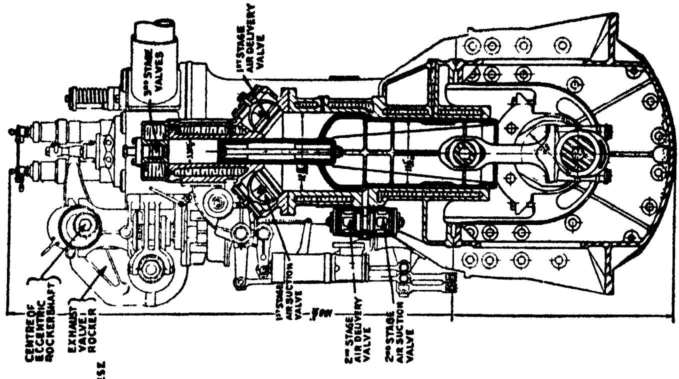


Fig. 248.—Cross Section Through One of the Air Compressor Cylinders

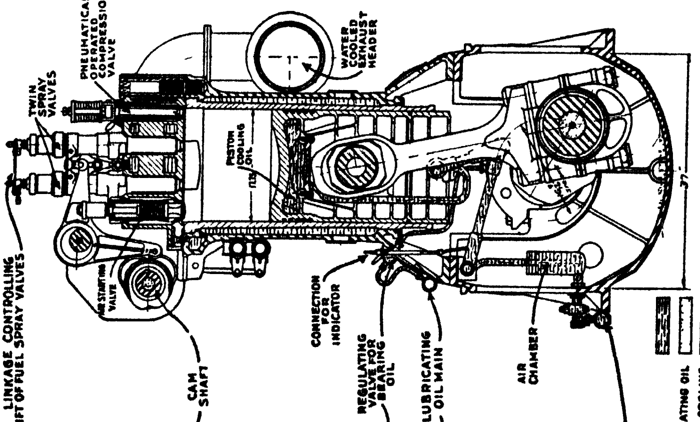


Fig. 247.—Cross Section Through a Main Cylinder

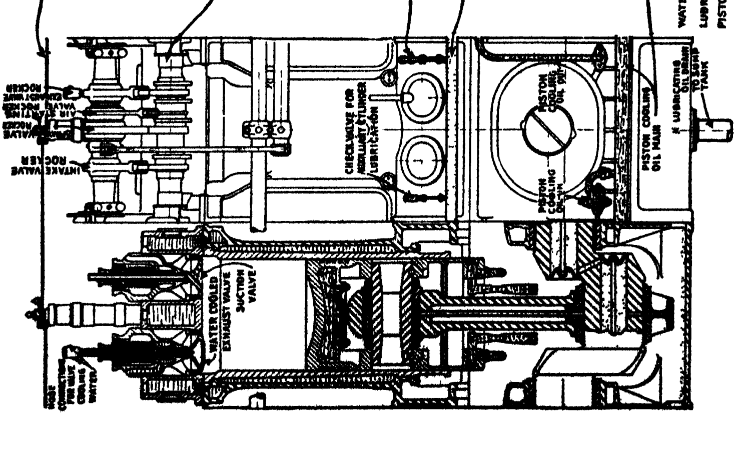


Fig. 246.—Longitudinal Section and Elevation Through Two Main Cylinders

greater while maneuvering, which is assurance that he will not neglect it. Probably the desire to avoid further complication from additional oil piping is the reason why these parts are not included in the circulating system. As it is, over 200 feet of oil piping are used, varying in size from ¼″ O. D. to 4¼″ O. D. Operators are instructed to oil by hand every hour the valve stems and all parts not included in the circulating system. A point of interest in regard to the valve lubrication is that the stems are intended to have sufficient clearance in their guides so that there is no contact at this point and therefore no oil is necessary. The only part lubricated is the piston-like guide above the spring, which is not exposed to heat.

A main oil sump tank of about 500 gallons is used for storage, and as it is located below the engine all the drainage of oil flows back to it by gravity, through outlets at both ends of the crank case. Operators are instructed not only to clean the tank every 8 or 10 weeks and the strainer contained in it every 50 hours of service, but to blow down the whole oiling system with steam once a year. From this tank the oil is drawn by either an auxiliary electrically-driven oil pump or the main engine driven oil pump. A "stop-lift" check valve on the suction line near the tank makes it unnecessary to prime the oil pumps when starting. Before starting the engine, complete lubrication is assured by the auxiliary pump which runs in parallel with the main oil pump. This auxiliary pump, shown in Fig. 249, is an interesting rotary type. It consists of a

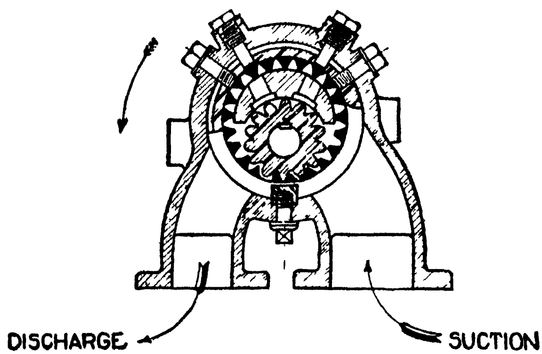


Fig. 249.—Electrically Driven Auxiliary Oil Pump

single spur gear mounted on ball bearings, and running in mesh with an internal gear. The outer diameter of the internal gear is turned down so as to break through to the space between the teeth, like a squirrel cage, giving passage for oil except for flanges at each end which hold the teeth together.

The main pump shown in Figs. 250 and 251, is a gear pump

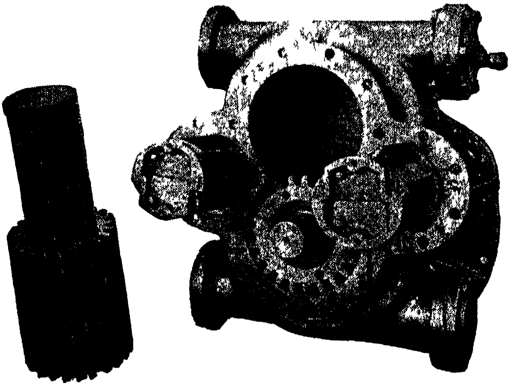


Fig. 250.—Direct Engine Driven Main Oil Pump Partly Dis-assembled

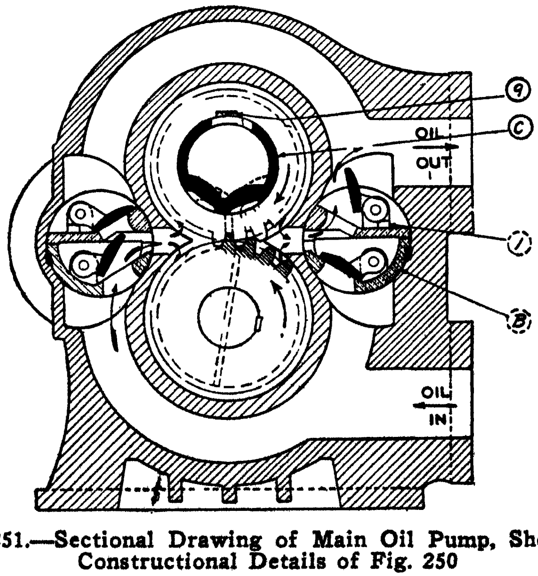


Fig. 251.—Sectional Drawing of Main Oil Pump, Showing Constructional Details of Fig. 250

driven at crankshaft speed by spur gearing from the vertical shaft which drives the camshaft, and operates with the gears in a horizontal plane. The two principal features of this pump are: (A) The means taken to release oil trapped between the teeth as they mesh, thus making it operate quietly, and (B) The check valves which allow oil to flow in one direction irrespective of the direction of pump rotation. The first feature is accomplished by means of nine 9-32″ holes drilled through the idler gear radially between the teeth. The spindle upon which this gear rotates is of comparatively large diameter, and has a series of transverse ports milled tangentially as shown in Fig. 250. Oil in the space between two teeth running into mesh can pass into these ports and back to the discharge side of the pump through similar holes between teeth which are still free. In a similar way the teeth going out of mesh are supplied with oil from the suction side. This results in eliminating the shocks and side pressure caused by teeth closing in on a trapped slug of oil—as well as the partial vacuum on the suction side. Thus the efficiency of the pump is raised, its life is prolonged, and the operation is made unusually quiet.

When the pump rotation is reversed by a reversal of the engine, a slight change is necessary in the location of the ports. This is automatically accomplished by the friction between the idler and its spindle, which tends to drag the spindle in the same direction that the gear rotates. A keyway in the pump housing, which is wider than the key in the spindle, allows a few degrees of rotation of the spindle when the gear motion is reversed, and thus secures the proper setting of the spindle ports. The Naval Officers who have had experience with the operation of this pump are enthusiastic in their praise of it.

From the pump, oil flows through a 2⅜″ I. D. pipe to the filter shown in Fig. 252. This is of the duplex type, and of such capacity that, by the three-way valve, either half may be by-passed for cleaning while the other is in use. In normal operation both sides are used in parallel. Instructions recommend that these filters be cleaned every 12 to 24 hours of service. Attention is called to the arrangement of the six

discs with gauze on each side, by which a large area is secured in a small space.

The filtered oil passes next to an oil cooler shown in Fig. 253. It will be noticed that the water tubes and the baffles for

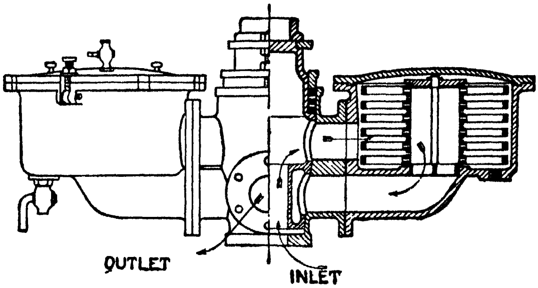


Fig. 252.—Duplex Oil Filter

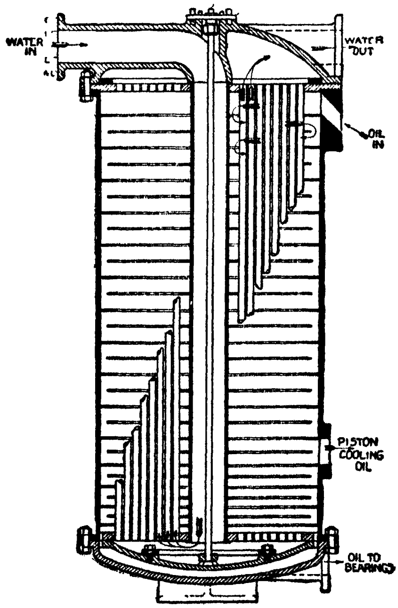


Fig. 253.—Lubricating and Piston Oil Cooler

the oil are suspended as a unit from the upper header, while the lower header is free, its vertical position being located by the tubes only. This results in two distinct advantages:

(1) The tubes are free to expand and contract without inducing stresses between themselves and the casing; (2) The tube unit can be removed readily from the casing, making cleaning and inspection a simple matter. Zinc rings are bolted to the top bronze header and the bottom iron cover to reduce corrosion from the salt water.

At the cooler, oil is divided into two separate systems, one for piston cooling and the other for bearing lubrication. The oil for the piston cooling system is drawn from a point 8⅜″ from the bottom, but the oil for the bearings is cooled still further by passing to the bottom between baffles which, on account of the reduced volume of oil, are spaced about ⅞″ apart instead of 1½″ as in the upper section. These two oiling systems will now be traced in detail.

Piston Cooling System. Oil for piston cooling passes from the cooler to a header along the base of the crank case, through air chambers and then to the pistons by pairs of hinged pipes, as can best be seen from Figs. 246 and 247. There are, of course, two of these oil connections to each piston, one leading the oil in and the other out. The purpose of air chambers is to cushion the "water-hammer" effect of the reciprocating mass of oil in the piston, which is severe enough to endanger tight oil joints.

The hinged oil pipes fulcrum on two pins at the lower end of each piston, one on each side. From these pins, which are hollow, oil is led by other piping to and from the closed-in pocket under the piston head as shown in Fig. 254. No precautions are taken to remove air from this pocket when starting, for in operation if any air is present the oil is splashed so violently that it is sure to be removed with the oil in a few minutes as a sort of froth. Oil from the pistons is led through 13/16″ I. D. tubes to a point near the operator's station, where the flow from each piston can be observed as it is collected in a funnel and returned to the sump tank. Thermometers are placed near each outlet, where the uniformity of the piston cooling can be observed. This oil cooling system is very satisfactory, and has a double advantage over use of water for the same purpose; there is no danger of incrustations of salt in the cooling system, and any leakage into the crank

case will not cause trouble with emulsions. It is true that a given rate of oil flow will carry off less than one-half as much

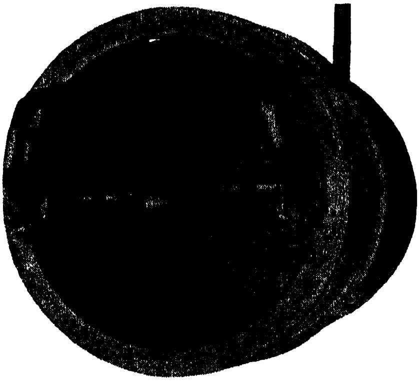


Fig. 254.—Interior of Main Piston Showing the Oil Piping to the Oil Pocket in the Piston Head, and Pin Joints Through Which Oil Connections Are Made to Stationary Parts of Engine

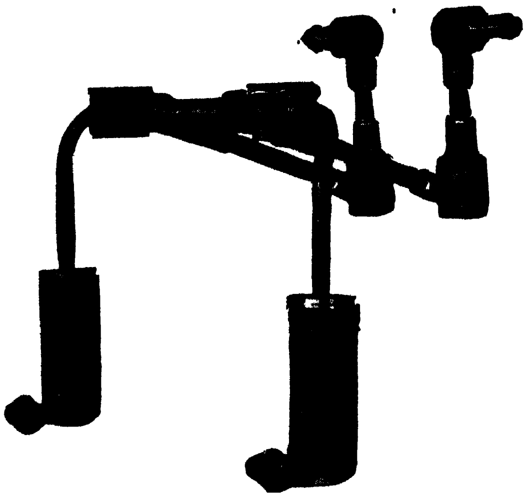


Fig. 255.—Hinged Piping Connections Between the Crankcase and Piston for the Piston Cooling Oil. Their Location in the Engine is Shown in Fig. 247

heat as water, but even in this engine all the heat is carried off that is necessary, with an oil temperature no higher than about 130° F. Approximately 280 B.T.U.'s per hour are car-

ried off per I. H. P. of the engine. It is interesting to compare this method of conducting oil to and from the pistons, with that of Fig. 256 which is used by Sulzer and many others.

Bearing Lubrication System. The second branch of the oil system from the cooler, as has been mentioned, is for lubrication. Oil is led through a 1½″ I. D. pipe to a pressure reducing valve shown in Fig. 257. Mounted with this reducing

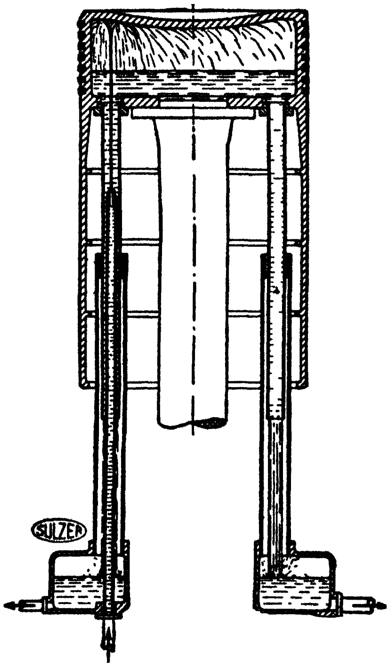


Fig. 256.—Piston Cooling System Used by Many Diesel Engine Manufacturers, in which Telescopic Tubes Are Used Instead of Hinged Joints

valve is a pressure relief valve which limits the pressure in the whole system back to the pump, and hence the piston cooling system also. From this reducing valve, oil is supplied to the lubricating oil header which runs along the crank case, from which ½″ I. D. leads pass to the center of each of the main crankshaft bearings at the top. Control over the flow to each bearing is secured by special stop cocks which can be locked

in any one of several positions intermediate between full open and shut, by means of a pin in the handle which may be dropped into various holes in a segment fixed to the valve body. Occasional touching of the bearings with the hand will

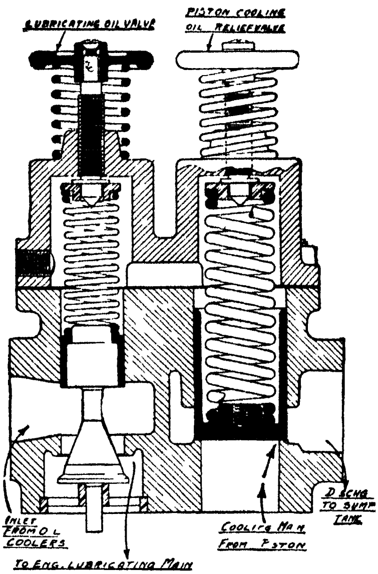


Fig. 257.—Oil Pressure Regulating and Relief Valve. By means of the Pressure Reducing Valve at the Left the Oil Pressure on the Bearing is Reduced to About 20 Pounds per Square Inch at 200 R.P.M., and Increases to a Maximum of 30 Pounds at 450 R.P.M. of the Engine. The Right Hand Member is Set to Protect the Oil System from Pressures of Over 50 Pounds

indicate by their temperature if they are running with too much or too little oil. A noteworthy feature of the crankshaft bearings is the absence of criss-cross oil grooves in the pressure surfaces. There is only one circumferential groove by means of which oil is led to drilled passages in the crankshaft and thus to the crankpins. All holes in the crankpins and bearings are threaded to take plugs when a shaft is disassembled for shipping, so that dirt cannot get inside.

From the crankpin bearing, oil is led up to the wrist pin bearing by a ½″ I. D. tube through the hollow connecting rod. The wrist pin bearing is a two part steel forging which, con-

trary to usual practice, is lined with babbitt and grooved as shown in Fig. 259. The wristpin, where it bears on the bearing, is of case-hardened steel, and each end is slightly tapered

Fig. 258.—Dis-assembled Connecting Rod Bearing, Showing the Type of Oil Grooving, which is 3/32″ Deep by ⅜″ Wide. The Relieved Areas at the Side Are 3″ by 6½″ Long, and .006″ Deep

Fig. 259.—Connecting Rod with Babbitt Lined Wrist Pin Bearing Showing the Very Small Amount of Grooving Used. There Are Four Crosses Like the One Shown, at 90 Degrees from Each Other and Connected by a Single Groove

so it can be fitted solidly in the piston. It is held in place by taper pins which are driven home, these being further locked by cotters, making a very simple and secure fastening. The assembly of this bearing in the connecting rod is unusual. The

lower half is first put in place, where it is secured by the flanges on each end when the upper half is pressed into place. To make this possible one of the flanges of the upper half is cut down lower than the thickness of the shim which goes between the rod and the top "brass," the exact thickness of which is selected to require a force of one ton to press this part into place.

Cylinder Lubrication. Oil thrown from the crankpin bearings is depended upon to supply the principal cylinder lubrication. An additional means is provided by leading oil directly from the lubricating oil header to points on opposite sides of each cylinder. A check valve on the line to each cylinder normally prevents any flow of oil, and is so located as to be convenient to the operator who presses each one open for a moment when the engine is started, or at any time when he may think one of the cylinders may need additional oil. In order to prevent too much oil from passing up into the combustion space, no other precaution is taken than the use of two special oil wiper rings, one of them being the lowest of the five rings at the upper end of the piston. These two rings which, like the others, are ½″ square in section, are beveled on the outside from the top to within ⅛″ of the bottom, producing a clearance of 3/64″ at the top. This concentrates the ring pressure on a width of ⅛″ when the ring is new. As the ring wears, the width of contact increases until it is ¼″, at which time the ring should be replaced by a new one. Below the grooves for these rings, the piston is chamferred for ½″ to a depth of 1/16″. Six ⅛″ holes are drilled through the piston walls at these chamferred spaces, to return the oil wiped from the cylinder wall. Drain holes are drilled also to ends of the wrist pin spaces.

The camshaft and valve rocker shaft have internal oil passages for their entire length, and are supplied by oil from the same header that feeds the main bearings and auxiliary cylinder lubricators. Holes through the rocker shaft where it passes through the rockers, supply oil to the latter and to the cam rollers by tubes which lead oil from the rockers to the hollow studs on which the rollers turn. The camshaft bearings get oil in a similar way from holes in the camshaft. The

two sets of spiral gears through which the camshaft is driven by a vertical shaft at the rear of the engine, are lubricated by streams of oil which strike the gears where they run together. The gears do not run submerged in oil. These oil streams are fed by the same header which supplies the main bearings, the camshaft bearings and the valve rockers and rollers. Thus all journals of the engine proper, which rotate while the engine is

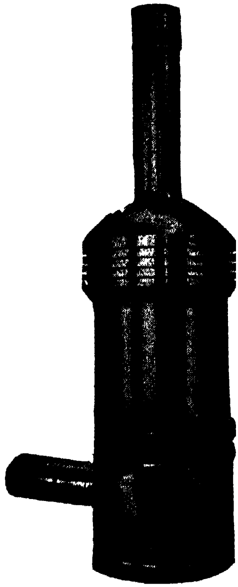


Fig. 260.—Air Compressor Piston, with Wrist Pin Partly Withdrawn

running, are supplied by a circulation of cooled oil under pressure, the pressure increasing with the engine speed.

Air Compressor. The air compressor has four stages arranged in a unique way, there being two "differential" pistons driven at 180° from each other by cranks on an extension of the engine crankshaft. Each of these pistons has a 1st and 2nd stage—operating in parallel. The aft piston also handles the 3rd stage, while the forward piston handles the 4th stage. Fig. 248 shows a section through the forward piston and its water jacketed cylinders, and Fig. 260 shows the piston itself. Notice that the bottom of the piston has to seal the 2nd stage

from leakage of air into the crank case. This minimizes the tendency for too much oil to work up into the cylinder from the crankpin, for there is not only no suction to draw it up but if there is any leakage of air it will tend to force the oil down from the cylinder walls.

This particular arrangement of the different stages on one piston, in which the 2nd stage acts on the down stroke while the 1st and 3rd act on the up stroke, instead of all three acting in one direction, not only reduces the bearing pressures but

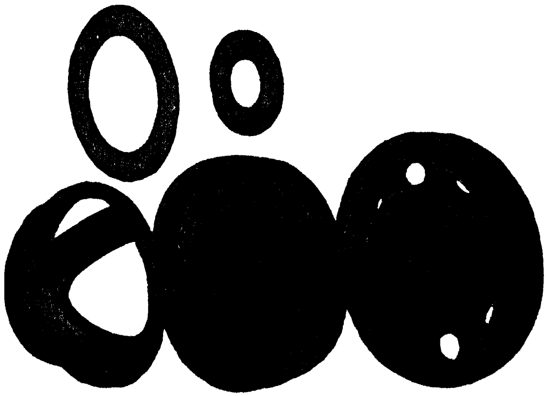


Fig. 261.—Automatic Suction Valve for First Stage. Valve Itself Consists of Two Thin Ground Spring Steel Rings

actually reverses them. It makes their lubrication much simpler than if the pressure were always in the same direction. Dividing the work of the first two stages between two cylinders in parallel not only insures a fairly steady stream of air, but still further reduces the total bearing pressure. The crankpin and wrist-pin bearings are lubricated from the hollow engine shaft by the same circulating system used for the other crankshaft bearings.

The cylinders, however, are lubricated by non-circulating mechanical oilers, the oil being used only once. There are two oilers, one having four plungers and the other two. There is one plunger for each stage of each cylinder. Each of these plungers in one stroke displaces from about 1 to $2\frac{1}{2}$ cubic inches of oil, but in operation over an hour is required to make

one stroke—a rachet, worm and screw thread being made use of to reduce the rate of the drive. This is practically all the oil the compressor gets in compressing 450 cubic feet of free air per minute to a pressure over 1,300 lbs. per square inch. Four hundred and fifty cubic feet is over double the quantity of air used in running, and the volume is normally reduced by

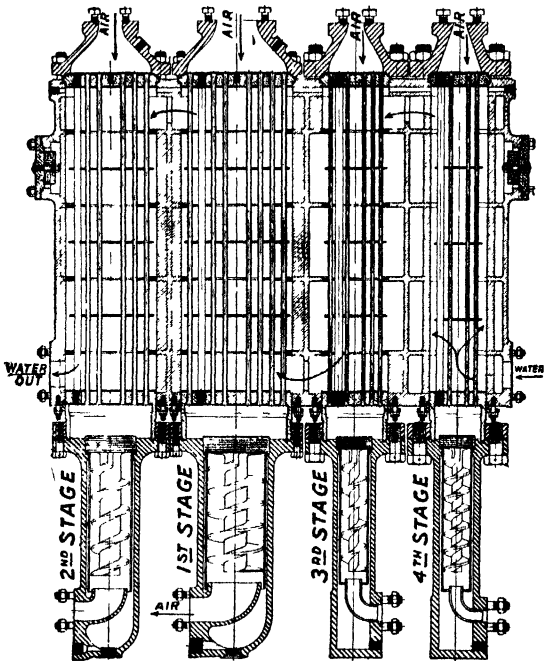


Fig. 262.—Air Compressor Intercoolers and Separators

throttling the suction to the 1st stage. The large reserve is necessary to fill the starting air bottles and for the ship's tanks. The valves are automatic, and, as can be seen in Fig. 261, consist of thin flat steel rings. The instructions are to clean the pistons and rings every six months, and the valves every 150 hours of service.

Particular precautions have been taken to separate out all water and any oil which are condensed in the intercoolers through which the air passes after leaving each stage. Much more water is separated out in the intercoolers than is com-

monly realized. With saturated air at 80° F. when compressing full capacity, over 5 gallons an hour would be separated. In normal operation this is probably about one gallon an hour. Fig. 262 shows a section of the air cooler and separator unit for all four stages. In the separators the air is whirled in passing along a helical passage and centrifugal force throws any liquid onto the wall of the surrounding tube. Liquid thus separated passes through narrow vertical slits in the tube and then drains down the outer wall into a retainer provided for the purpose below the air outlet. Every 15 minutes of operation these retainers are "blown down" to remove the condensate which has collected. Notice that a nest of cooling tubes and a separator forms a unit suspended from the top header only, and leaves the tubes free to expand and contract and in this respect is similar to the oil cooler. The lower header of the cooler is made water tight by a gland packed with a rubber ring. These separators are simple and effective, and are an important item in the reliable operation of the compressor. Passage of water into the compressor cylinders greatly interferes with their lubrication, and thus the reliability of the whole unit.

Fuel System. The fuel used in submarine Diesel engines in the U. S. Navy has a viscosity normally of less than 100″ Furol at 77° F., though oil of 1,000″ at the same temperature (about 16° Be. gravity) has been burned so satisfactorily when preheated that it is believed still heavier fuel can be used in these engines. The only apparent drawback was that the maximum power developed without a smoky exhaust was a little lower, possibly because the fuel was not so thoroughly atomized. These engines give a clean exhaust over a surprisingly wide range of speed and load, even at a mere idling speed. In order to accomplish this extreme flexibility several so-called mechanical "complications" have been added, notably means for varying the spray-air pressure and the spray valve lift with the engine power. These features can be called "complications," only in a service where their advantages would be unnecessary, such as for low speed freighters. The fuel system, being at least as novel as some details of the lubricating systems, merits special consideration.

Fuel oil is displaced from the main tanks to a service tank by the admission of sea water. An electric device indicates when the water level reaches a pre-determined height so that the engine can be switched over to another tank, or stopped

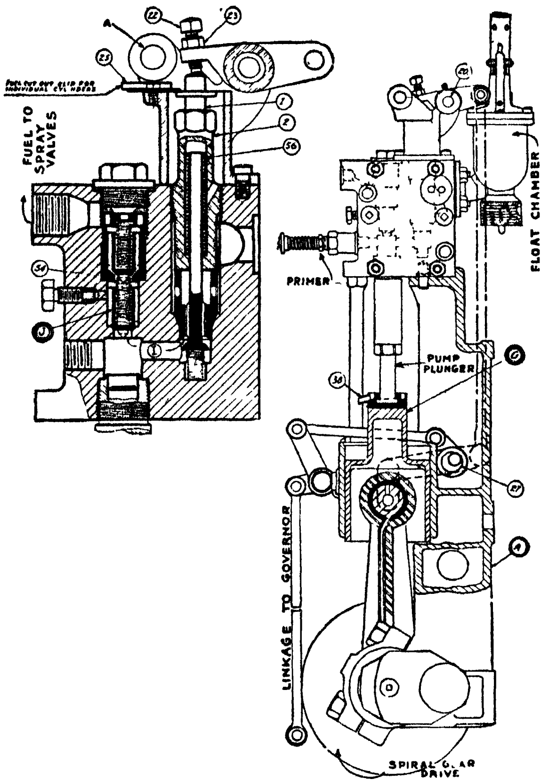


Fig. 263.—Cross Section of Fuel Pump Drawing to Twice the Scale of the Side Section Shown at the Right. Plungers are 11/16" in Diameter and Have a Stroke of 5⅛". Regulation of the Quantity of Fuel Fed is Secured by Holding the Automatic Suction Valves Off Their Seats for Part of the Delivery Stroke, by Means of the Linkage Shown. The Eccentric Fulcrum (27) Operated by the Throttle or Overspeed Governor, Regulates the Extent of Plunger Travel When the Suction Valve (1) is Allowed to Seat. Individual Regulation of Fuel for Equal Power in Each Cylinder May Be Adjusted by the Tappet (22). When the Engine Control Levers Are in the "Stop" Position All Delivery is Stopped by the Rotation of a Rod at "A" Which Opens All Suction Valves by Short Arms Which Bear Down on the Collars of (1). Any Cylinder Can Be Cut Out of Service by a Clip (25) Which, When Set as Shown, Holds the Suction Valve Permanently Off Its Seat

before salt water is fed to the fuel spray valves. This is not only a mechanically simple method, but eliminates any appreciable change in ballast.

The service tank is large enough to contain about two hours' supply of fuel and is provided with muslin filter cloths. These are very satisfactory but must be renewed at least every six weeks, and oftener with some grades of fuel. The tank, as well as all fuel oil leads, are recommended to be emptied and carefully cleaned with steam twice a year if possible, as is good practice with any fuel oil system. From the service tank the fuel flows by gravity through a $1\frac{1}{8}''$ I. D. pipe, to a float chamber similar to but larger than those used for automobile carburetors. This float is located just back of the fuel pump, as shown in Fig. 263, and is provided with a sight glass to show the presence of fuel.

The fuel pump unit consists of two groups of four individual pumps, the plungers of each group being driven at 180° from the other. The pump runs at half engine speed, of course. Three of the plungers in one group deliver fuel to spray valves while the fourth pumps lubricating oil to the spray air regulator. The three fuel plungers deliver simultaneously to cylinders 1, 5 and 4, while three of the other group deliver to cylinders 6, 2 and 3. The pump is so timed that the delivery stroke of the 1, 5, 4 group is just completed when the piston of No. 6 main engine cylinder is on its firing dead center. All plungers are in a row, are hardened, ground and lapped in their guides so that no stuffing boxes are used, and are loosely coupled to their respective cross-heads so that they will not bind even if not in exact alignment. The small amount of leakage past the plungers, is caught in a trough and drained so that it will not make the engine look untidy. Two check valves in tandem are used for the discharge valves from each pump to securely withstand the very high pressure of the injection air. The quantity of fuel delivered to each cylinder, and thus the engine torque, is regulated in the usual Diesel way by holding the suction valves open for a greater or less part of the stroke. Accurate control of the fuel to each cylinder is very important, because the quantity of fuel burned on one firing stroke is very small. Even in an engine of this

size the maximum delivery on a single stroke will be but 0.2 cubic inch which may be visualized as a sphere less than $\frac{3}{4}''$ in diameter.

For the purpose of filling the fuel lines before starting, there is a priming plunger (16) for each fuel pump. When

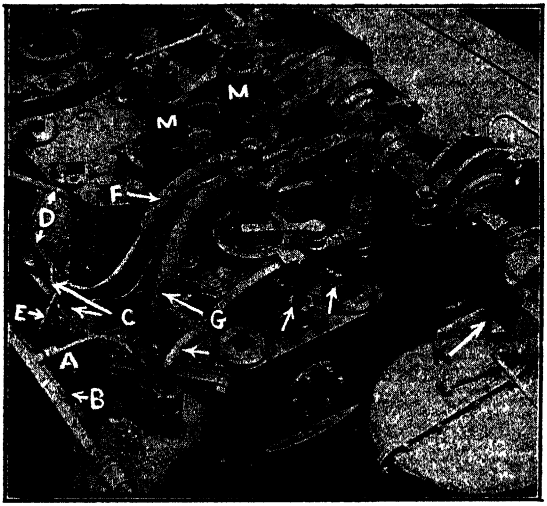


Fig. 264.—View Looking Down on Rear Cylinder Head Showing the Arrangement of Parts and Rubber Hose Connections Which Supply Water Circulation to Exhaust Valves

F—Water in to Exhaust Valve
G—Water from Exhaust Valve
H—Water to Exhaust Valve Cage
J—Water from Cage.

All the Cylinder Cooling Water Passes Through These Two Paths in Parallel and Then Into the Water Jacket Around the Exhaust Pipe

D—Spray Air Lines to Spray Valves M.
C—Separate Fuel Lines to Each Spray Valve from Test Screw A
B—Single Fuel Line from Fuel Pump to Test Screw A
K—Lubricating Oil Lead to Valve Rocker Shaft
L—Lubricating Oil Lead to Spiral Cam-Shaft Gears
E—Air Line to Cylinder Compression Relief Valve.

it is operated it makes use of the same valves that are normally used by the regular plunger, and consequently when priming, the throttle must be set in its full open position and the interlock to the starting lever must be disengaged so that the suction valves of the fuel pump will not be held off their

seats. When priming, little "test-screws," shown at A' in Fig. 264 near the spray valves are opened and fuel is pumped by the primer until it flows out. Drain tubes are provided to carry off the fuel bled out. A check valve is used between these test-screws and the spray valve to prevent passage of spray air when the screw is open. When the fuel lines are filled to these screws, the engine is ready to start as far as the fuel is concerned.

The fourth pump of the fuel pump unit previously mentioned is almost identical to the other pumps. It delivers lubricating oil to the spray air regulator valve at the same rate that fuel is pumped to the cylinders. This regulator, shown in Fig. 265, automatically reduces air pressure from the air flasks to a pressure varying from 600 lbs. per square inch at no load to 1,175 lbs. at maximum load. The more fuel pumped to each cylinder, the higher will the spray air pressure be. If the engine were suddenly stopped while running at full load, the spray air lines would remain filled with air at the maximum pressure. In order to prevent the existence of such a pressure when starting again, a simple spring loaded relief valve is opened by the control levers when moved to the stop position. This valve is set to blow down the lines to 600 lbs., so the proper pressure for starting is secured automatically.

Fuel Injection Valves. One of the most interesting features of this engine is the use of two spray valves for each cylinder. This is probably done in order to allow larger inlet and exhaust valves than would be possible if space were occupied in the center of the cylinder head by the spray valve and to permit more uniform water circulation. At the same time good combustion is augmented by injecting fuel at two points rather than one. The general arrangement of the twin valves is shown in Fig. 247, together with the single rocker by which they are opened simultaneously. Greater detail is given in Fig. 266.

The fuel supply for each cylinder is divided at the test-screw (A of Fig. 264) into a line for each of the two spray valves. Equal division of the fuel is accomplished by: (1) Forcing the oil into each line through a small hole which causes considerable pressure drop; (2) The combined areas of

the divided lines are equal to the single supply line; (3) The lines to the valves are equal in length.

When running at low speed the actual time a fuel valve is open is ordinarily greater than at full speed, for the cam holds it open a given number of degrees of crankshaft rotation. The result is that at low speed unless some precautions are taken

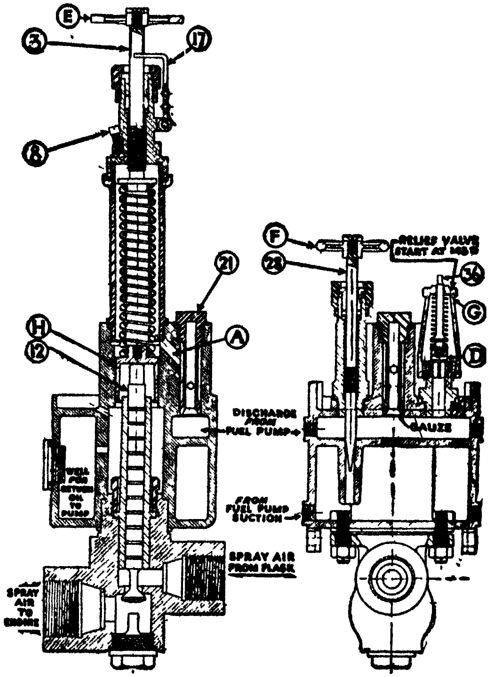


Fig. 265.—Spray Air Regulator, Which Reduces Air from the Pressure of the Air Bottles to a Pressure Depending on Quantity of Fuel Delivered to Each Cylinder. The Valve (12) is Balanced Against the Supply Air Pressure. Spring Normally Holds it Open, but Pressure of Air to the Spray Valve Acts Against It, Tending to Close the Valve. Hence the Stronger the Spring Pressure the Higher Will Be the Reduced Air Pressure. Spring Pressure is Adjusted by the Hand Wheel E for an Air Pressure of 600 Pounds When the Engine is Not Running. Lubricating Oil from the Two Extra Plungers of the Fuel Pump Enters the Space Around the Spring by Passing Through a Gauze Strainer (21) and Port A. Exerting a Pressure Against Piston H Augments the Spring Pressure and Thus Raises the Spray Air Pressure. Control of the Oil Pressure on H is Effected by the By-pass Valve (28), all By-passed Oil Returning to the Reservoir, Shown in the Casing, from Which it Flows Back to the Pump. A Safety Valve at the Right Protects the System Against Excessive Pressure If Valve (28) Is Accidently Closed Too Far

to the contrary, much more spray air enters the cylinder than is necessary for atomization, which is not only wasteful of the air supply but tends to chill the cylinder charge appreciably. Since great flexibility is needed by these engines, the opening

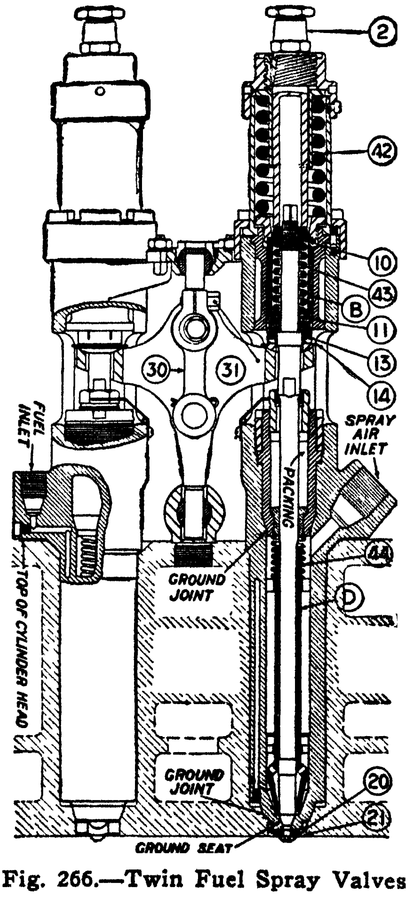


Fig. 266.—Twin Fuel Spray Valves

of the fuel valve is changed by varying its lift, in addition to varying the spray air pressure.

Engine Control. Engine control is very simple and almost fool-proof, one hand-wheel ("throttle") automatically regulating not only the quantity of fuel delivered to each cylinder and the lift of the spray valve, but indirectly varying the

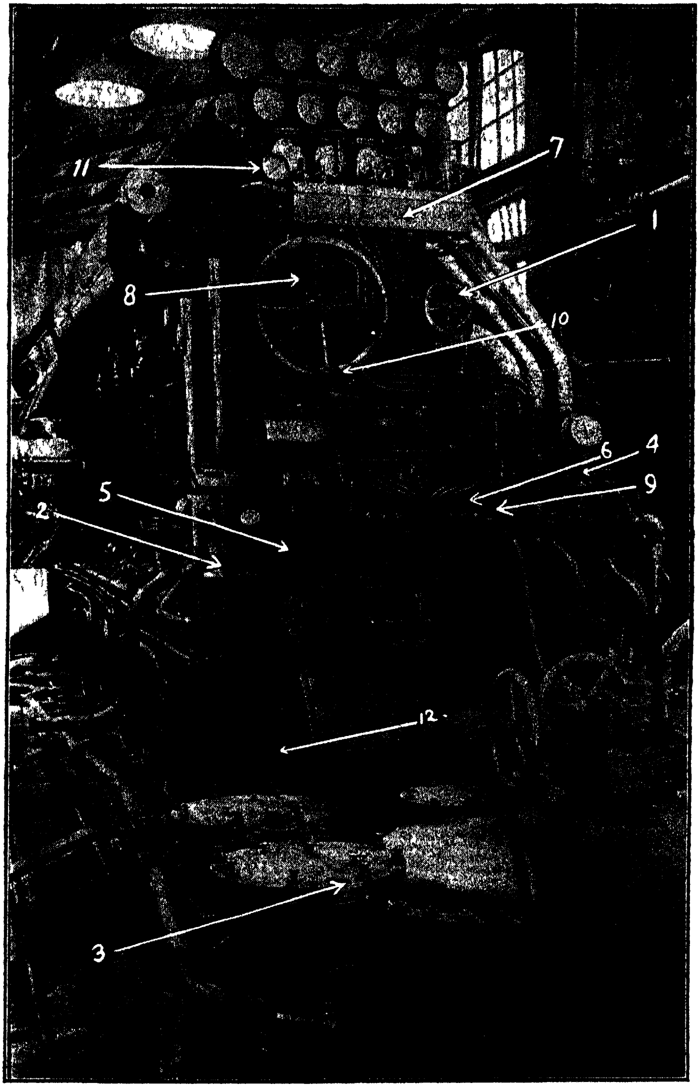


Fig. 267.—Control of the Engine, Showing Location of Many of the Parts Described in This Chapter. At the Left Are the Two Starting Levers; Just to the Right of Them is the Reverse Wheel (10) and the "Throttle" Control (1). In the Submarine a Platform is Located in a Suitable Position for the Operator to Reach These Controls, and Observe the Many Gauges Above the Engine. In Foreground Are Shown the Duplex Oil Filter (3) of Fig. 252, and Immediately to the Right is the Top of the Oil Cooler of Fig. 253. The Lubricating Oil Valve of Fig. 257 is Shown at (4); Fuel Pump Unit of Fig. 263 at (6); Spray Air Pressure Reducing and Regulating Valve of Fig. 265 at (9); and (7) Shows Grouping of Valves by Which the Air Compressor Intercoolers Are Blown Down and Connections Made for Charging Bottles, etc.

spray air pressure also. To provide complete control there are in addition only the two starting levers and a large hand-wheel for reverse. The large hand-wheel moves the camshaft to positions for ahead or astern by a method similar to that of Workspoor except that the eccentric fulcrums for the valve rockers are not tilted, and the camshaft is slid axially to change cams, like that of Burmeister & Wain, and Worthington. In all intermediate positions of this wheel, air operated compression release valves are opened for all cylinders. It is interlocked with the starting levers so that it can be moved only when the latter are set for "stop." On larger engines of this type the reverse mechanism is hydraulically operated and requires but two seconds for complete motion. The longer of the two starting levers when pulled back carries the shorter one with it, and by eccentric fulcrums disengages the spray valve rockers and brings the air starting valve rocker in contact with its cam. Further movement back opens the pilot valve throttle, throwing starting air onto the starting valves. When the engine makes about 100 r.p.m. the long lever is pushed all the way forward, removing air starting valve rockers for cylinders 4, 5 and 6 from their cams, and returning the spray valve rockers for these cylinders so that they can start firing. Cylinders 1, 2 and 3 are still air driven, and after a few more revolutions they are switched over also, by pushing the short lever forward. In stopping, the long lever is pulled half-way back and draws the other with it, shutting off all fuel irrespective of the throttle position, and blows down the spray air lines to 600 lbs. The flow of cooling water from the engine pumps, of course, starts and stops with the engine. There is hence no possibility that the operator can incorrectly handle the controls, nor run the engine with any adjustment of spray air pressure or spray valve lift except that for which it was intended. No miss-move can be made even if he "loses his head."

When starting the engine, time must be given for parts to warm up before putting on heavy load. Operators are instructed when starting an engine from cold, not to come to normal cruising speed until it has been running at least five minutes; to full speed, for 15 minutes; nor to emergency speed

for 20 minutes. After stopping, circulation of the lubricating system must be continued by the electric auxiliary oil pump for 15 minutes to cool off the pistons. When a long layup is planned the engine should be turned over at about 200 r.p.m. for a quarter of an hour with all lubricating devices in operation to assure all moving parts being thoroughly oiled. All external parts are to be cleaned and thickly covered with grease to prevent rusting. The cleaning, greasing and motoring should be repeated every month, and in addition—as is customary with engines—the crankshaft should be jacked over by hand once a day, care being taken to stop with the piston in different positions each time.

CHAPTER XIII

Diesel-Electric Drive for Ships

Westinghouse Electric and Mfg. Co.'s System of Diesel-Electric Drive

Selection of Engines—Description of Units—Main Driving Motors—Generators—Electrical Wiring—Description of 2500 S.H.P. Diesel-Electric Drive—Switches and Relays—Getting Underway—Reversing—Stopping—Securing—Port Operation—Control—Advantages—Weight—Reliability—Space—Fuel Consumption—Operating Costs—Tankers—Yachts—Tugs—Ferries—Passenger Ships—Present and Future Developments.

Although the Diesel-electric system of ship propulsion is relatively new, the constituent parts making up the system are well established. A Diesel-electric system preferably consists of two or more Diesel engine driven generators furnishing power to a motor driven propeller. The ship may be of the single, twin or multiple screw type. By using two or more generating units for a propeller, definite advantages in the way of weight, flexibility, control, reliability, etc., as discussed are readily obtained. The simplicity of the Diesel-electric system is obvious when it is realized that the principal component parts comprise only four pieces of apparatus, such as Diesel engines, generators, motors and control, two of which are quite similar.

Selection of Power. In selecting the power for a Diesel-electric system, we have a choice between alternating-current and direct-current. For the reason that direct-current obviates operating difficulties ensuing from alternating-current parallel operation; eliminates changes in engine speed; adds enormously to the simplicity, refinement and economy of control; and provides greater power in case of casualty to a generating unit, the direct-current system is obviously the proper system to use. In cases where a single generator supplies power to a single motor, alternating-current could be used

without encountering difficulties ensuing from parallel operation, but such a system would be far inferior to the direct-current system in flexibility, reserve power and control. Furthermore, a multiplicity of generating units, as are used in the case of d-c. systems, adds considerably to the ultimate reliability. Therefore, in the large majority of cases, Diesel-electric systems will utilize direct-current.

In the case of direct-current systems there are two general arrangements of machine connections that suggest themselves. One arrangement is to operate the generators in parallel, and control the motor speed and maneuvering by armature rheostatic means. This system, however, is rather cumbersome, wasteful during maneuvers and speed changes, and necessitates a complicated controller. The other employs what is known as the voltage-control, or Ward-Leonard control system. With this system, pure shunt machines are used and both motors and generators are separately excited, preferably from the same source. The motor fields are excited at constant potential, and always in the same direction. The excitation of the generator fields is varied to suit the motor speed and direction of rotation desired. By varying the voltage applied to the armature terminals of a shunt motor, having a constant field excitation, the motor speed can be varied in direct proportion, both as regards speed value and speed direction; and since the voltage generated by a constant speed, separately excited, shunt wound generator is directly proportional to its field excitation (neglecting saturation), the motor speed is, in turn, proportional to the generator excitation. With such an arrangement, therefore, it is only required to vary the generator fields from full excitation in one direction to full excitation in the opposite direction, to cause the motor to maneuver from full speed ahead to full speed astern. To further simplify this method of control, all machines are connected in series. With the series connection, it is unnecessary to maintain like speeds on all the engines. Provided the generators are excited equal amounts and have identical performance, the only effect of difference in engine speeds is a proportional difference in the loads carried by the generators, and their driving engines. From an operating standpoint, there-

fore, the series system is ideal, and permits by far the simplest system. Parallel operation of generators with the Ward-Leonard system would be very difficult, in fact, almost impractical.

Since it is necessary to handle only the generator field excitation currents for maneuvering the ship from full speed ahead to full speed astern, or holding any particular desired speed, the economy of the Ward-Leonard system is obviously superior to that of the armature rheostatic system during any other than full speed operation, for the reason that the generator field excitation power does not exceed 1½% of the total output of the generator. Dealing with these small currents, the control is extremely simple and inexpensive. This simplicity has a further direct effect on the maintenance of the equipment.

Brief Description of Units. Engines: The engine used with Diesel-electric propulsion may be any reliable make of Diesel engine, which operates at a reasonably high rotative speed, such a one being shown in Fig. 268. The term "rotative speed" is used instead of "speed" to distinguish from high piston speeds. Many people associate the engines used with Diesel-electric propulsion with those used for submarine propulsion. Diesel engines which are properly designed for use with Diesel-electric propulsion need not exceed established safe piston speeds for continuous operating engines. By using many cylinders of short stroke and small bore, the heat stresses common to large cylinder, slow speed engines are minimized, and the result should be an engine requiring less maintenance, and an engine of simpler construction.

By resorting to higher rotative speed engines, it is well known that the weight per brake horsepower can be brought down very rapidly. This characteristic is an important one in connection with Diesel-electric drive, as the amount of weight thus saved in the engine is considerably more than that added by the electrical machinery, and hence results in a total machinery installation which is lighter than that of any other type. It is confidently anticipated that the weight of a Diesel-electric propulsive installation using properly designed engines, should be approximately ½ that of a twin screw

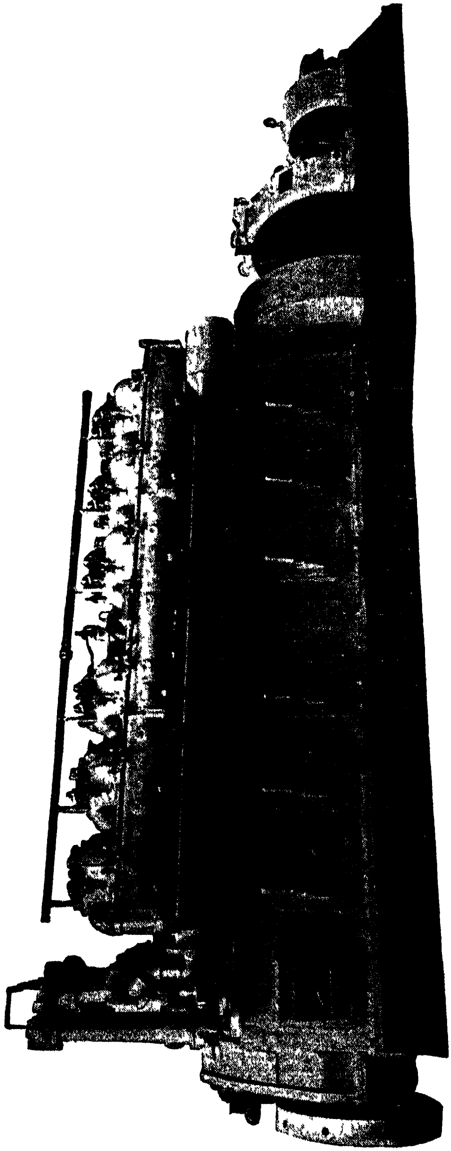


Fig. 268.—Arrangement of a Diesel Engine Generator Set

direct-connected Diesel propulsive system, and in the neighborhood of 75% of the weight of an economical, geared-turbine propulsive equipment.

Generators. The generators used with the preferable form of Diesel-electric propulsion are simple, direct-current, shunt machines, the construction and performance of which are easily comprehended by any person having a mechanical turn of mind. These machines consist essentially of two parts, the

Fig. 269.—Field of Typical Direct Current Shunt Motor or Generator

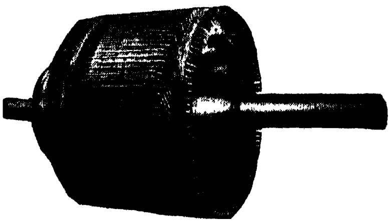


Fig. 270.—Typical Armature for Direct Current Shunt Wound Motor or Generator

field or stationary element, Fig. 269, and the armature or rotating element, Fig. 270.

The field is made up of a cylindrical steel ring, split at the horizontal center line for convenience, and having an elliptical section. Electrically, this frame serves to carry the field flux, and mechanically to support the field poles. The frame is machined on the inside diameter in order to form a true seat for the field poles which are bolted to it and symmetrically spaced. The main field poles are composed of a number of die-punched laminations of sheet iron, which are riveted together

to form a solid pole. The commutating field poles are built of solid steel and located between the main poles.

The main field coils which produce the field flux consist of a large number of turns of insulated copper wire having a relatively small section. The coil is wound on a form, slipped on the field poles before they are bolted to the frame, and rigidly supported from these field poles by insulated supports. These coils are known as shunt coils, and are connected in series.

The winding for the commutating field pole consists of a relatively small number of turns of bare copper strap secured by insulated supports. This winding is connected in series with the armature, and carries the line current. The purpose of the commutating pole winding is to provide a magnetic field to neutralize the effects of the current reversal in the armature coils undergoing commutation, and thus to effect sparkless commutation. Since the commutating field winding is in series with the armature, and carries the same current, the correct amount of commutating pole flux is automatically provided under all conditions of load within the capacity of the machine.

The armature consists essentially of a cylindrical core built up of steel laminations which are dovetailed and secured to a cast spider, the spider in turn being pressed and keyed on to the shaft. The steel laminations are provided with teeth punched in their periphery, and into which the armature coils are placed.

The commutator to which the armature coils are connected is made up of a series of hard drawn copper bars securely insulated from one another by means of mica insulation. These commutator bars are built up on a separate spider and securely fastened thereto by means of "V" rings fitting into insulated machined recesses in the bars, or by some other suitable means. The commutator spider is then pressed on an extension of the armature spider, or directly on the shaft and keyed thereto.

The armature coils, Figs. 271, 272, 273, are form wound and completely insulated and treated so as to be moisture resistant, before they are placed in the slots, and connected

to their respective commutator bars. The armature coils for any given machine are identical.

The armature is usually carried on a forged steel shaft having an integral flange at one end which is bolted directly

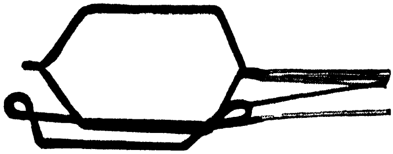


Fig. 271—Complete Armature Coils

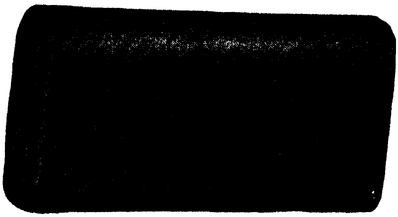


Fig. 272.—Section Through Armature Coils

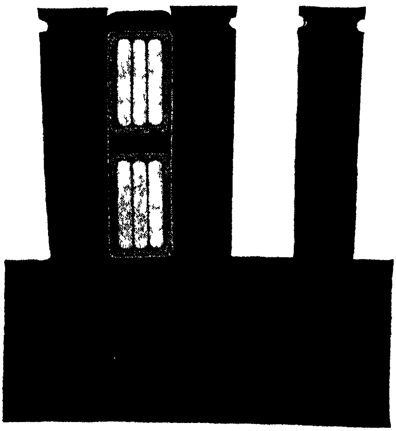


Fig. 273.—Section Through Coils and Slots

to the flywheel of the engine, the other end being carried by a pedestal type bearing.

The brush rigging, Fig. 275, which properly constitutes a part of the stationary member, serves to collect the current

from the commutator, and is supported from the field frame. There are the same number of brush arms as main field poles. Brush arms are symmetrically placed and so located that the brushes rest on commutator bars which connect to armature coils, which lie in the commutating zone, which in a commutating pole machine is midway between the main field poles.

The brush arms carry a series of brush-holders, each of which is provided with a carbon brush connected to the brush-

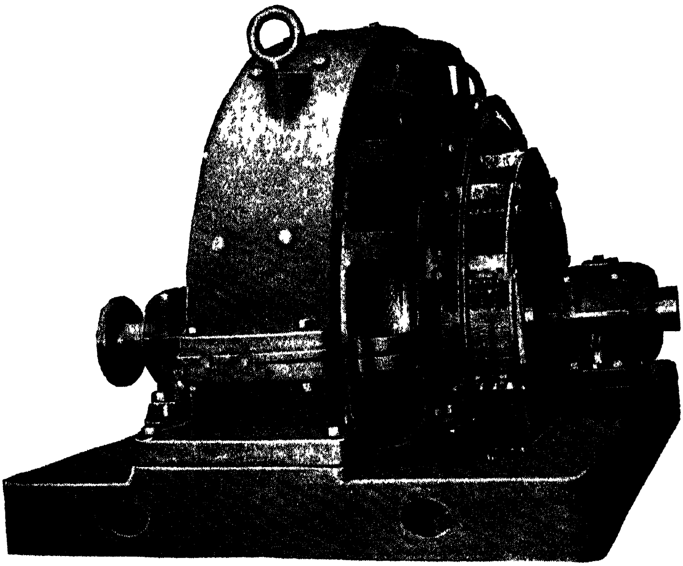


Fig. 274.—Main Driving Motor

holder by means of a copper shunt (sometimes called a pigtail). To assure an equal distribution of current in the brushes, an adjustable spring is provided on each brush holder which maintains the pressure of the carbon brush on the commutator at a given predetermined correct value.

Motors. The corresponding description of the motor, Fig. 274, is identically the same as that of the generator, and therefore, will not be given. In order to minimize the total weight, it is preferable to provide the motor without a bed-

plate, and simply to provide feet on the field frames and bearing pedestals suitable for mounting on a built-up structural steel bedplate in the ship. The structure supporting the motor should be rigid so as to avoid distortion.

The bearings of the generators are usually supplied with lubrication from the engine lubricating system. In the case of the motors, it is usual to provide oil ring lubrication. In some cases, however, forced or flood lubrication is provided.

Exciter Arrangements. The exciters for the generator and motor fields may either be driven by the main engines, or by separate engines. In either case, they may furnish power to the auxiliaries in addition to that for excitation.

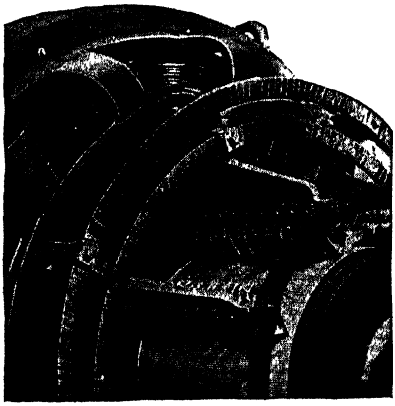


Fig. 275.—Method of Supporting Brushes

If driven by the main engines, the exciters should be direct-connected to the main generator either by means of a coupling, or by mounting on an extension of the generator shaft. In the latter arrangement, the exciter armature may be overhung if the mechanical factors permit. Driving the exciters from the main generator shafts by means of chains, belts or gears, effects a slight saving in weight and overall length of the set; however, it is not nearly as satisfactory mechanically as direct drive.

Whether it is best to use direct driven, or separately driven exciters, depends upon such factors as sea load, desired flexi-

bility, capacity of main generators as related to port demands, available space, etc., and each case must be considered on its merits. When the arrangement is convenient, it is usually preferable, however, to drive the exciters by the main engines, as it results in a self-contained propulsive plant.

Control. The control consists of a suitable switchboard containing the necessary switches for the several machines

Fig. 276.—Pilot House Aboard the "J. W. Van Dyke." This is a Typical Pilot House for a Diesel-Electric Ship

involved, the instruments, protective relays, circuit-breaker, etc.; a special reversing field rheostat for the generator field circuits; and a manually operated, remote control mechanism, preferably mounted on a pedestal for operating the field rheostat.

There are two general methods of operating the field rheostat. One method employs a handle which operates in the fore and aft direction, and is thus similar to present steam engine

control. This handle operates the rheostat through a system of rods and bevel gears. The other system employs a worm and wheel instead of the handle.

The worm and wheel method has an advantage, in that an inherent time element is introduced in the control mechanism.

This time element is essential, as too rapid change in the field strength would cause serious overloads on the machinery. It has been found from actual service that the minimum time which would be consumed in bringing the propellers from full speed ahead to the stopped condition is approximately 5 seconds, and by designing the worm and wheel so that it would require 5 seconds to make the number of turns necessary for full speed to stop position, this required time element is automatically provided.

However with the lever type pedestal, a time element in the form of a magnetic drag, or copper disc rotating in a magnetic field, may be incorporated in the pedestal, so that it really matters little whether the worm and gear or the lever type pedestal is used. Either type may be made for either single or twin screw.

The switches for the machines are so arranged that any particular generator or motor unit may be taken out of service by simply throwing its switch from one position to another. This operation is usually effected without interrupting the service to the propeller motor.

A 2500 S. H.P. Diesel-Electric Drive. To convey a clear idea of Diesel-electric drive, it is thought best to describe a specific case. The example selected is a 2500 S. H.P., single screw drive, having four 500 K.W., Diesel driven, 250 volt, generators supplying power to a 2500 H.P., double unit, 90 R.P.M. motor. Two 75 K.W. Diesel engine auxiliary generating sets are provided for supplying the excitation and auxiliary load while at sea. The following is the list of apparatus constituting the drive:

1—2500 H.P., 90 R.P.M., double unit, direct-current, 500-volt, shunt motor. The two armatures are mounted on a forged, flanged shaft carried by two pedestal bearings. The motor frames and the bearing housings and bearings are split along the horizontal center line to provide easy access.

4—500 K.W., 250-volt, direct-current, shunt generators, direct coupled to four Diesel engines. The generators arma-

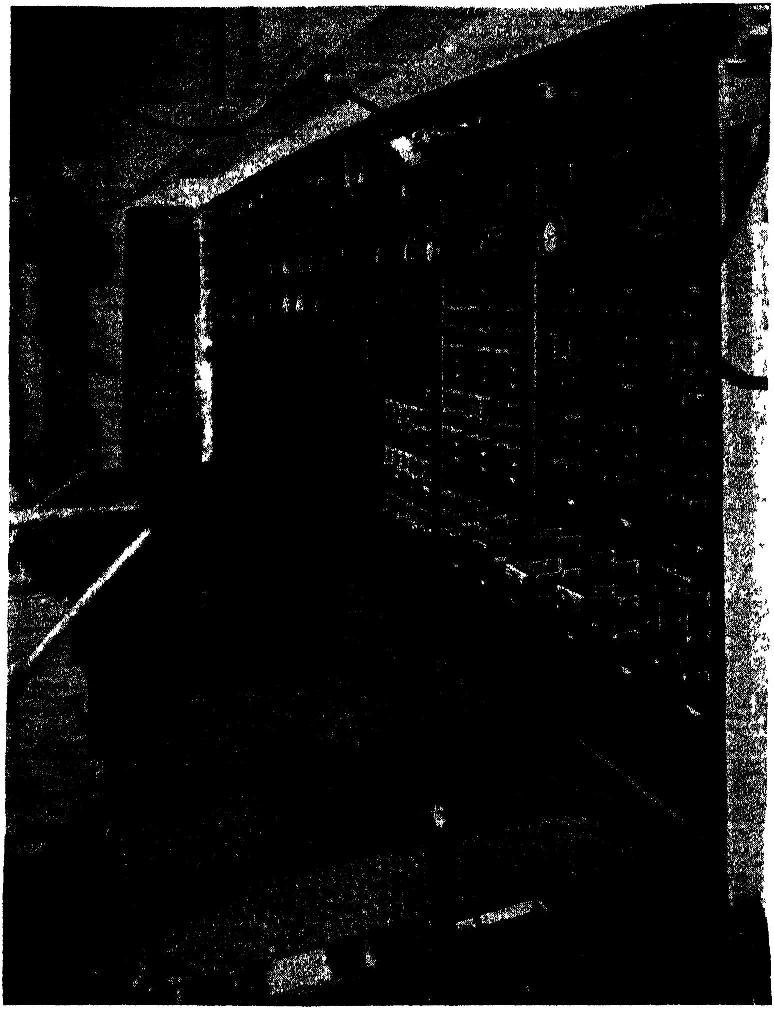


Fig. 277.—Switchboard of the "J. W. Van Dyke." Dead Front Feature on Heavy Current Panels

ture is mounted on a forged, flanged shaft supported at the commutator end by a pedestal bearing and coupled to the

engine flywheel at the rear end. As in the case of the motors, the frame and bearings are split.

2—75 K.W., 250-volt, compound wound, direct-current, Diesel engine driven, auxiliary generators. The mechanical arrangement is the same as that of the main generators. One of these sets serves as a spare.

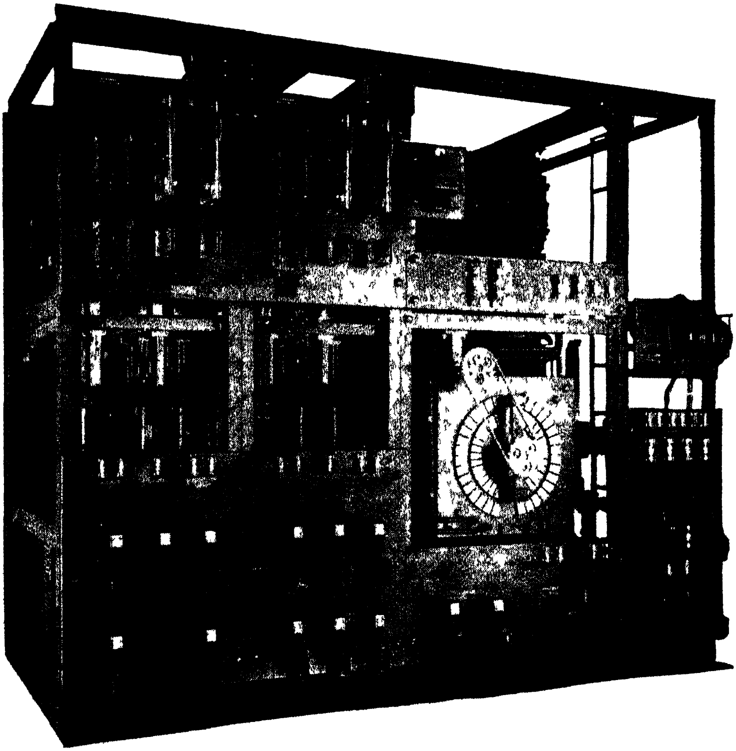


Fig. 278.—Rear View of a Dead Front Switchboard for a Tug Boat. Control Pedestal for Operating Field Rheostat is Located in Pilot House

1—Complement of motor driven, engine room auxiliaries, such as oil pumps, circulating pumps, auxiliary air compressor, sanitary fresh water, fire and bilge pumps, etc.

1—Switchboard and control for the above machinery.

The four main Diesel generating sets are located forward; and the motor, auxiliary Diesel generating sets, vise bench,

switchboard and control station are located aft. The oil supply tanks and other accessories are located on the upper grating. The location of the control station is such that the operator has full view of the propelling machinery, and hence the operator can observe the performance at all times. The engines are arranged right and left hand, so that their controls and gauge boards may be conveniently handled and observed. The entire arrangement provides accessibility and convenience for operation and inspection, and at the same time is not wasteful of space.

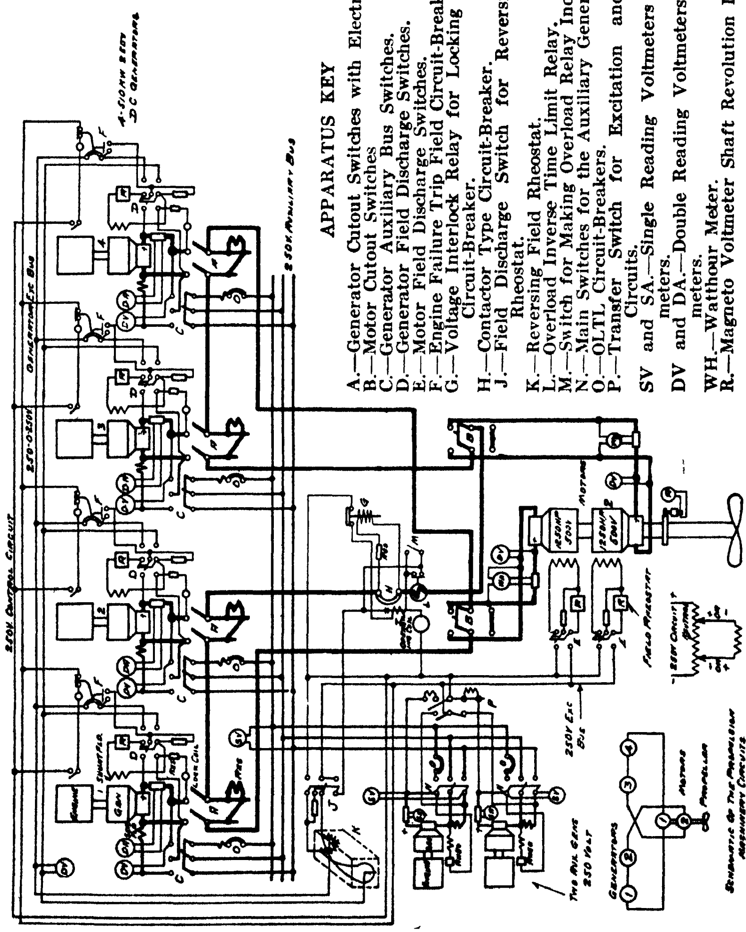
Switchboard and Control. The small diagram at the lower left-hand corner of the following control diagram, Fig. 279, shows the scheme of connections, and the main diagram shows the full details of connections, including all necessary switches, circuit-breakers, relays, etc. A glance will show that all main machines are connected in series, and, that the motor and generator armatures are interspersed so that the current passes through the circuit in the following order: Generator No. 1, Generator No. 2, Motor No. 2, Generator No. 3, Generator No. 4, Motor No. 1, and back to Generator No. 1, thus constituting a series circuit.

Since the total voltage of a chain of series connected generators is the sum of the voltages of the individual machines, the total effective voltage in this case is 4x250 or 1000 volts. However, by interspersing the motor armatures in the manner stated, the ground voltage, or the maximum voltage between any two points in the system is only 500 volts. Such an arrangement is advantageous, in that the circuit is really a 1000-volt system from a current standpoint, but only a 500-volt system from a voltage or insulation standpoint. In other words, it necessitates only one-half the copper that would be required in a 500-volt system of this same capacity, and at the same time does not exceed the 500-volt insulation strain.

Switches and Relays. The principal switches, relays, etc., are designated by letters, and all switches performing the same function bear the same letter. For instance, all generator cutout switches are designated by the letter "A."

"A"—Two-pole, manually-operated, transfer, knife cutout switch for generators. When closed in the upper position,

these switches connect their respective generators in the propulsion circuit; and when thrown to their full lower position,



APPARATUS KEY

A.—Generator Cutout Switches with Electric Lock.
B.—Motor Cutout Switches
C.—Generator Auxiliary Bus Switches.
D.—Generator Field Discharge Switches.
E.—Motor Field Discharge Switches.
F.—Engine Failure Trip Field Circuit-Breaker.
G.—Voltage Interlock Relay for Locking Out Main Circuit-Breaker.
H.—Contactor Type Circuit-Breaker.
J.—Field Discharge Switch for Reversing Field Rheostat.
K.—Reversing Field Rheostat.
L.—Overload Inverse Time Limit Relay.
M.—Switch for Making Overload Relay Inoperative.
N.—Main Switches for the Auxiliary Generators.
O.—OLTL Circuit-Breakers.
P.—Transfer Switch for Excitation and Control Circuits.
SV and SA.—Single Reading Voltmeters and Ammeters.
DV and DA.—Double Reading Voltmeters and Ammeters.
WH.—Watthour Meter.
R.—Magneto Voltmeter Shaft Revolution Indicator.

Fig. 279.—Control Diagram for a 2,500 S.H.P. Diesel-Electric Drive

their blades connect to a solid bar, thus cutting the generator out of the propulsion circuit and establishing the series propulsion circuit through the bar between the lower contact

jaws. The upper and lower portions of the blades are at an angle, so that the switch makes contact on the first set of lower jaws before breaking contact on the upper jaws, and vice versa. In the lower throw, the right hand lower blade engages two jaws in sequence. The first jaw inserts a resistance which prevents a rush of current, due to the residual field of the generator. Further closing breaks the connection to the generator armature on the upper jaws, and engages the bar on the lower jaws. This switch is electrically locked against being thrown to the lower position until the generator field has been opened.

"B"—Two-pole, double-throw, motor cutout, manually-operated knife switch. This switch has no special features as it is not operated when the circuit is alive. The upper position connects the motor in the propulsion circuit, and the lower position cuts out the motor and establishes the propulsion circuit through the bar between the lower jaws.

"C"—Three pole, single throw, main generator auxiliary bus switches. These switches are provided in order to utilize the main generators when in port for supplying auxiliary power. It will be noted that this switch is three pole to permit parallel operation (equalizer connection), and that a series field is connected in the circuit to give the generator the desired compound characteristics.

Switches "A" and "C" are interlocked so that only one or the other can be closed in the upper position at the same time. This prevents using the generator for two purposes.

"D"—Generator field switches.

"E"—Motor field switches.

"F"—Engine failure trip, field circuit-breaker. These are provided to make the generator automatically ineffective and to prevent its motorizing in the event of failure of its engine. This breaker is connected in the separate excitation circuit only as such protection is unnecessary when the generators are operating on the auxiliary bus. The means for opening this field circuit-breaker is actuated by a mechanical attachment on the engine, or by a voltage differential relay; the latter, however, is rather complicated.

"G"—Voltage balance relay for assuring that the motor counter voltage and the generator voltage are approximately at the same value before the automatic main circuit-breaker can be closed. This lock-out feature is necessary in case the main circuit-breaker trips while the ship is under way.

The device consists essentially of a spring-closed relay contact in the auxiliary circuit of the closing coil of the main automatic circuit-breaker, and a polarized magnet, one pole of which is excited by the generator voltage and the other pole of which is excited by the motor voltage. When the two voltages are equal, the flux produced in the magnet core by the two windings is neutralized and there is no pull on the relay arm, and the auxiliary circuit to the main circuit-breaker coil remains intact. If the generator voltage is appreciably different from the motor voltage, a pull is exerted on the relay arm by the polarized magnet, and the relay contact is opened, thereby preventing the main circuit-breaker from closing.

"H"—Automatic reclosing circuit-breaker located in the propulsion circuit. The function of this device is to protect the machinery against practically short-circuit conditions. The breaker is provided with an inverse time element overload relay. In the event of very severe or sustained dangerous overloads, this relay, whose magnet is excited by the main current, will open the circuit of the circuit-breaker closing coil through the auxiliary relay, and disrupt the main circuit. To again close the circuit-breaker, it is necessary to adjust the generator voltage by means of the main control handle, to the value of the motor counter voltage. To re-establish the proper generator voltage, it is merely necessary to move the control handle slowly to the "increase" or "decrease" position, as the case may be, and when the proper position is reached, the breaker will close automatically.

"J"—Field discharge switch for the reversing rheostat.

"K"—Reversing field rheostat, Fig. 280, for main generator field excitation when generators are connected to the propulsion circuit.

(Note—When the main generators are used for auxiliary power, they are self-excited and operate as normal compound-wound generators.)

The simple diagrammatic scheme of the type of field rheostat used for the control of the propulsive machinery is shown near the lower left hand corner of the control diagram. The rheostat is constantly energized from the excitation circuit. The leads to the field slide symmetrically over buttons on the rheostat face plate which are connected to the resistance at regular intervals. The arrangement is such that the lead contacts of the field circuit effectually cross each other at the

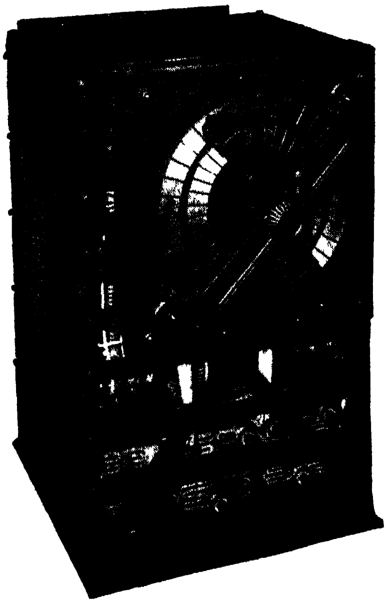


Fig. 280.—Typical Reversing Rheostat

middle point of the rheostat in going from full excitation in one direction to full excitation in the other, and hence not even a field circuit is opened in going from ahead to astern.

"L"—Overload inverse time limit relay. This is described under "H."

"M"—Switch for making overload relay inoperative. The purpose of this is to provide a means for making the circuit-breaker non-automatic in the event that maneuvers in dangerous or restricted waters make it imperative to maintain a

positive couple between the motors and generators. To make the relay inoperative, it is merely necessary for the operator to press a button, or close a small switch which bridges the overload relay contacts.

"N"—Three pole, single throw, knife switches for connecting the auxiliary generators to the main bus. Three pole switches are provided to permit parallel operation.

"O"—Overload time limit circuit-breakers for all generators when connected to the auxiliary bus.

"P"—Two pole, transfer switch for excitation circuits. This switch is very similar to "A," except that both throws are like the lower throw of "A" and have the preventive resistance.

The purpose of the switch is to provide a ready means for quickly transferring the excitation circuits from one auxiliary generator to the other, and also to provide an excitation connection to the auxiliary generators which is unprotected by circuit disrupting devices.

Preparing to Get Under Way. Upon receipt of notice to prepare for getting under way at full power, the first operation is to see that all switches are in the proper position. Close switches "A" and "B" in upper position; close switch "D" to the right; close field circuit-breakers "F"; close excitation switch "P" to the exciter which is in operation; see that generator field control handle is in the "off" position, and start the engines.

Getting Under Way-Ahead. Upon signal to get under way, close the field switches "E" and "J." (The closing of "J" establishes power to the circuit-breaker closing coil.) Move control handle ahead in answer to the signal and adjust the speed to the required value.

Getting Under Way-Astern. Proceed as under way "ahead," as described above, except move the control handle to the astern direction.

Stopping. If it is merely desired to stop the ship without regard to time, move the control handle slowly to the "off" position.

If it is required to stop the ship quickly, as in the case of

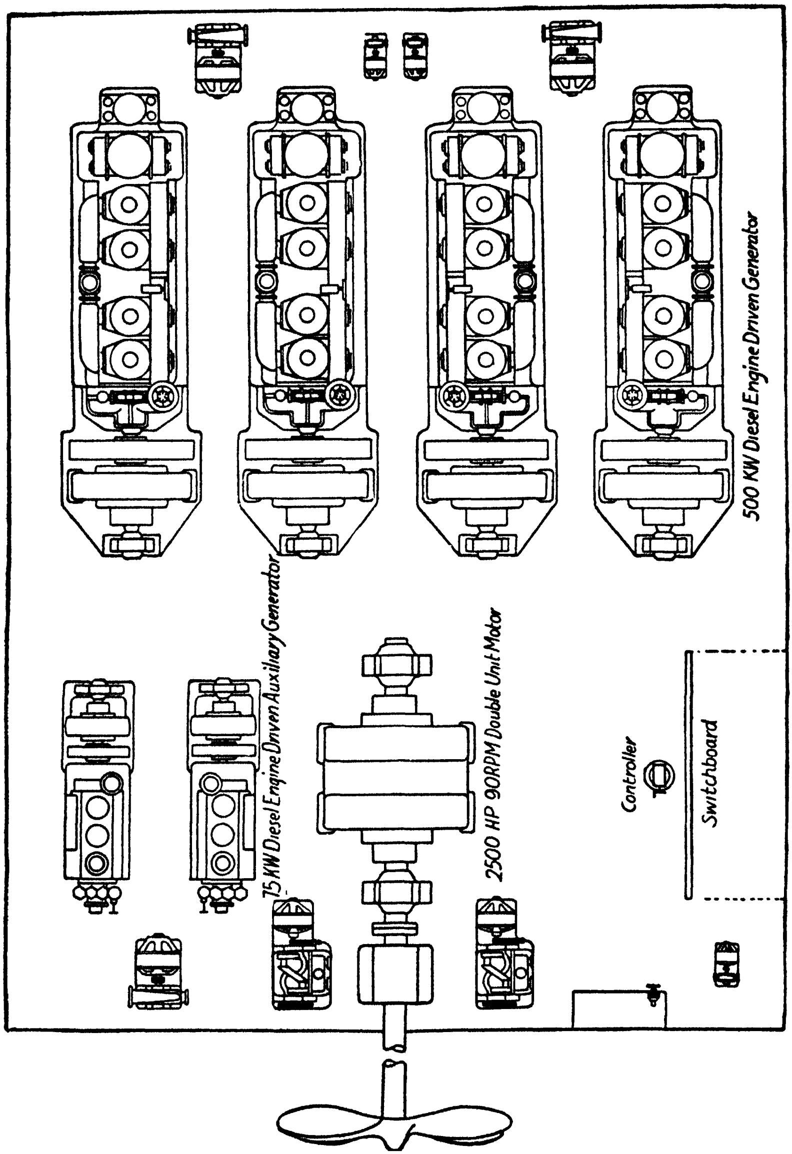


Fig. 281.—Plan View of Engine Room for a 2,500 S.H.P. Diesel-Electric Drive

an emergency, move the control handle toward the "off" position at such a rate as will maintain the current at 100 per cent to 150 per cent of full load value. In this operation, the cur-

rent will reverse as soon as the generator voltage is reduced below the motor counter voltage, and will stay reversed until astern operation ceases.

Special Set Ups. If the machinery is idle when changing set ups, the switches may be thrown without any special precautions. If, however, the ship is under way at the time of

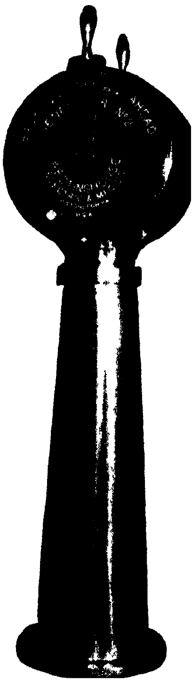


Fig. 282.—Generator Field Control Pedestal for a Drive Having Two Motors Independently Controlled. All Maneuvers of the Ship Are Effected by Motion of the Small Levers

changing set ups, reduce the voltage by means of the individual field rheostat to approximately zero, and trip the field circuit-breaker of the generator which is to be taken out of service. Then throw switch "A" to its extreme lower position after the electric lock described above has released the switch lever. Then shut down the engine.

To put a generator back into service while the ship is under way, reverse the above operations.

Since the probability of having to take a motor unit out of service while the ship is under way, is extremely remote, no provision is made for disconnecting it without interrupting the circuit. To take a motor unit out of circuit, move the main generator field control slowly to the "off" position; open

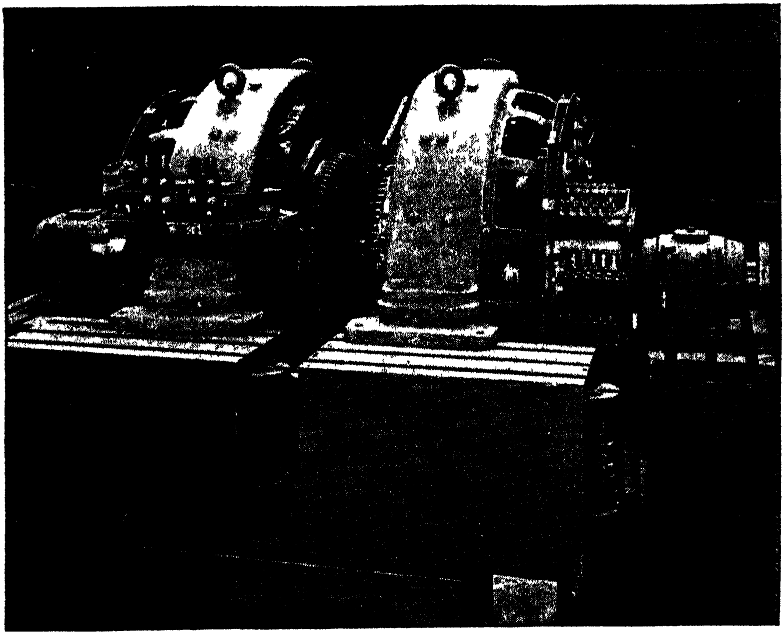


Fig. 283.—Typical Double Unit Direct Current Shunt Wound Propulsion Motor on Test

the generator field rheostat switch "J" and the proper motor field switch "E"; then throw the proper switch "B" to the lower position. To re-establish the power, close the switch "J" and operate the control handle in the normal manner. When operating with one motor, it is unnecessary to use more than two of the generators.

Other set ups must be performed in a similar manner.

Securing Electrical Machinery While in Port. When the ship has docked, close all generator switches "A" in the full lower position, open motor switches "B," open generator field

switches "D" and generator field breakers "F," open generator field rheostat switch "J" and open motor field switches "E."

If the machines are to be idle for more than 24 to 48 hours, and it is expected that they will be subject to appreciable changes in temperature, or will cool noticeably below the ambient air, their fields should be excited at a low value to prevent sweating (condensation of moisture on the windings). A separate circuit (not shown in the diagram) is usually provided for this purpose. Although the insulation is made as moisture resistant as practice permits, the windings should not be allowed to sweat or become wet.

Port Operation. (A) To use the main generators in port, for cargo winches and auxiliary power purposes, the following operations are necessary:

1. Start the engines which are to be used.
2. Close field switch "D" to the left for self-excitation.
3. Close circuit breaker "O."
4. Build up voltage to normal by cutting out resistance of field rheostat. (If the voltage builds up in the wrong direction, open switch "D," and close it to the right, close switch "J" and build voltage up in proper direction to about half value, by moving the main control handle.) Open "J" and close "D" to the left, and repeat as above.
5. Close switch "C," thus connecting the generator on auxiliary bus. Adjust voltage.

(B) To parallel a second main generator:

1, 2, 3, 4. Same as above.

6. Be sure that the machine voltage is the same as the bus voltage, and then close switch "C."
7. Adjust voltage so that the machines divide the load as indicated on the ammeters.

(C) Subsequent machines are paralleled in the same manner as described under "B."

(D) If one of the auxiliary sets is in operation when the first main generator is connected to the auxiliary bus, it must be paralleled as in "B."

(E) The auxiliary sets are paralleled with each other and with the main generators on the auxiliary bus as described in "B."

Stopping and Reversing. Because of the inherent functioning of a Ward-Leonard system of Diesel-electric propulsion it is necessary to give some thought to the inherent characteristics of propeller performance, particularly quick stopping when under full headway. Turning warrants consideration only in the case of multiple-screw ships, and only then in particular types of drive, such as turbine-electric using alternating-current machinery.

During quick stopping, however, it is necessary to overcome the propeller torque in order to bring the propeller to

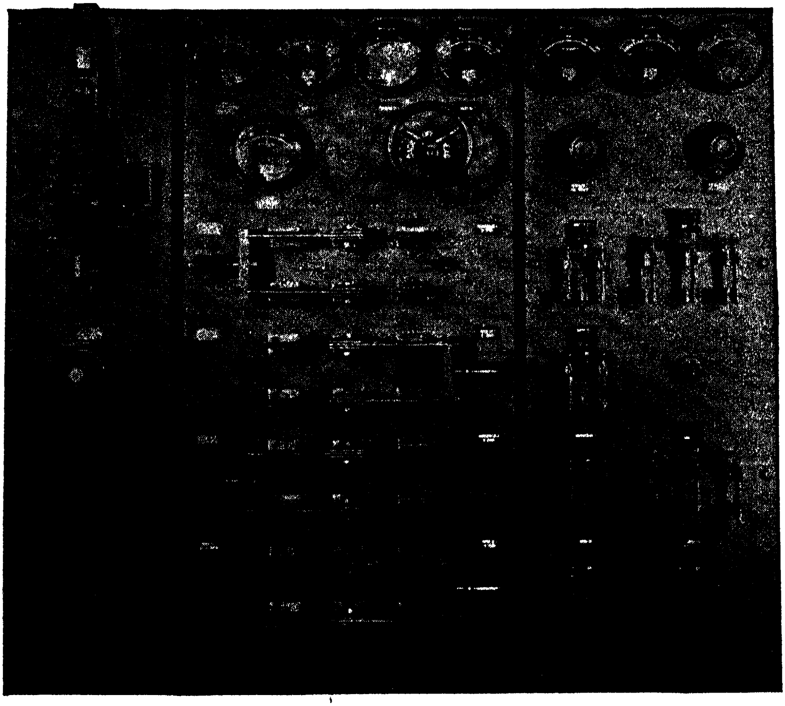


Fig. 284.—Switchboard and Control Panel for Small Diesel-Electric Drive, Consisting of Two Generators and One Double Unit Motor Operating on the Principle Described. Generator Field Control is Mounted on the Left Panel. Switches for Cutting In and Out the Various Generators and Motor Units Are Mounted in the Center Panel. In This Particular Case the Switches Are Not of the Transfer Type, and Switching Must Be Done on a Dead Circuit

rest. This propeller torque is developed by reason of the motion of the ship through the water, which causes the propeller to be driven as a water motor. Although strict analysis of this performance is not pertinent to the present discussion, it is well to recognize the results. In the case of a pure Ward-

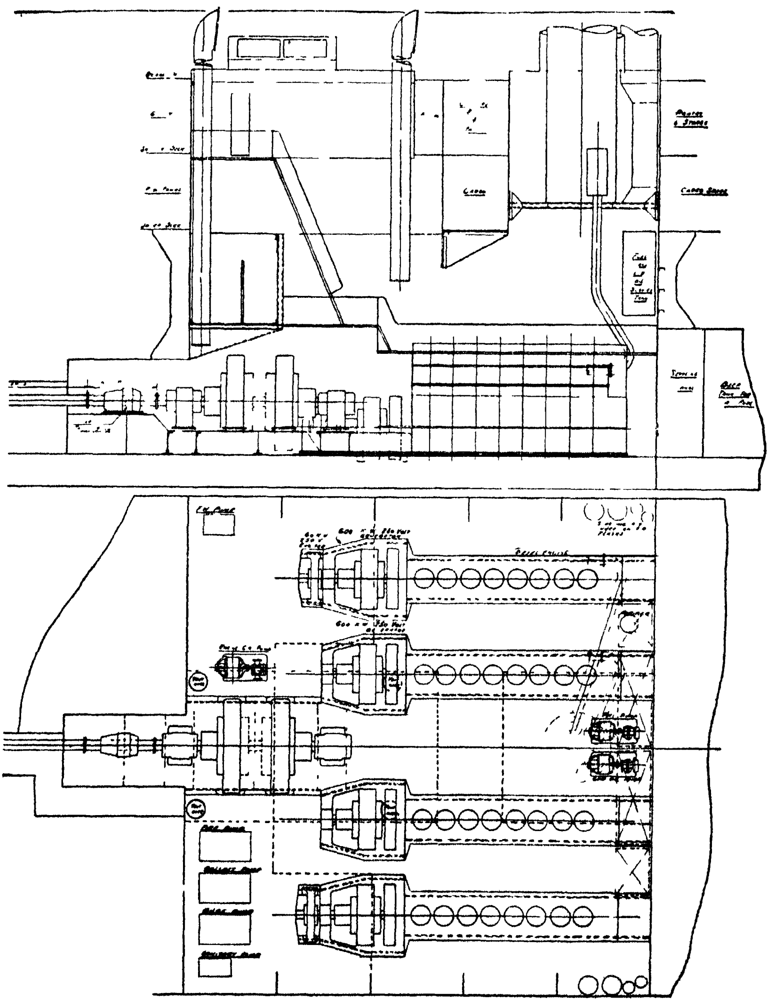


Fig. 285

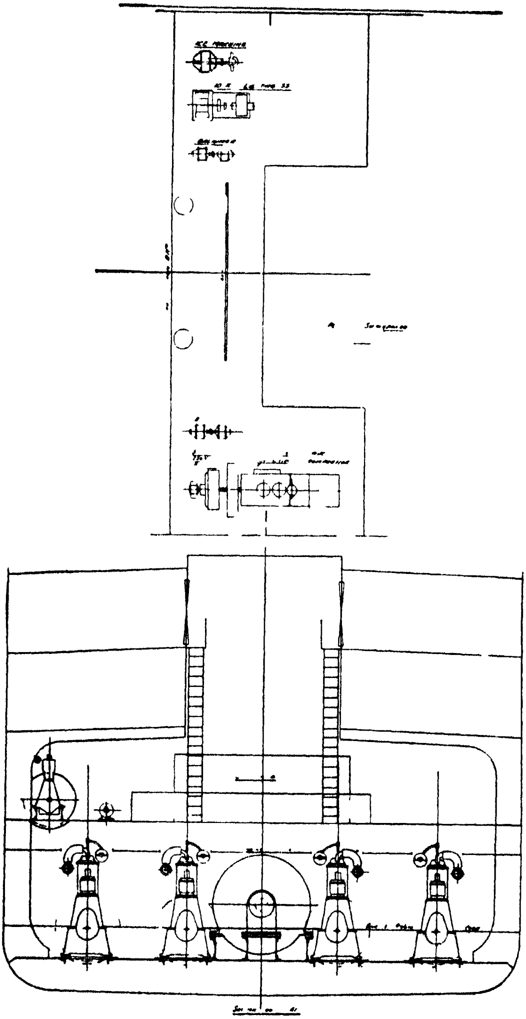


Fig. 285.—Schematic Arrangement of Engine Room for Diesel-Electric Drive of a 9,400 D.W.T. Shipping Board Vessel Design 1015, with Dimensions L.O.A. of 402 ft. 6 in.; Beam, 53 ft.; Draft, 26 ft. 6 in. Suggested Propelling Equipment Consists of Four 850 B.H.P. Eight-Cylinder Diesel-Engines, Driving Generators and Exciters at 250 R.P.M., Generating Power for a 2,800 S.Ph. Motor Running at 90 R.P.M and Ship's Auxiliaries

Leonard system of Diesel-electric propulsion, the principal means of absorbing the energy returned through the screw is the friction of the engines. If more energy is returned than can be dissipated by the friction losses in the engines, and the losses of transmission, external means such as dynamic braking resistors must be provided. If all the energy returned by the screw is not dissipated by the frictional losses of the engines, or otherwise it will be expended in increasing the speed of the engines. Whether or not the increased speed would be detrimental to the engines can only be conjectured. However, it is confidently believed that in the large majority of cases, and particularly those of the ordinary cargo ship, that this energy will not be in excess of that which can be absorbed by the frictional loss of the engine. Specific tests of this performance were made in the case of some Diesel-electric yachts when stopping from full speed, and it was found that there were no increases in the engine speeds due to this cause.

A conservative analysis shows that the maximum horsepower returned to the engines when bringing the propellers to rest from full speed, is approximately 33%, and that this peak value will last for a very brief instant only. The average horsepower returned is approximately 18%. The values take into account the torque produced by the propeller and the inertia forces of the motor armatures and the propeller, based on a stop of five seconds. Since the average four-cycle engine with attached auxiliaries is approximately 75% efficient, and the average two-cycle engine with attached auxiliaries is approximately 72% efficient, based on brake horsepower, it is apparent that the frictional losses in either case amount to at least 33% of the engine output. There is a slight margin in these figures, as no allowance was made for the propeller shaft bearing friction. Due to this returned energy when making quick stops, or reversals, it is necessary to design the engine governors to throttle to practically zero oil flow, in order that the friction losses of the engine may provide a load for absorbing the returned energy.

In cases where the compressors, etc., are separately driven, and in cases where the returned energy otherwise is in excess

of what the engines can absorb, it is necessary to connect resistance in the circuit during quick stops, for absorbing the excess. This is very easily arranged, and its insertion is done automatically by an auxiliary circuit actuated by virtue of relative position or motion of the control device.

Turning. When turning at full power, there is an increase in the load on the propellers. This increase is particularly pronounced in the case of multiple screw ships, as the power builds up enormously, especially on the inboard side of the turn if means are not taken to guard against it. This characteristic causes some concern in the case of alternating-current drives, and necessitates special control devices. Even though the input to the prime mover be limited to normal running value, there is a drop in its speed and an increase in torque of approximately 25% to 30%. Since a normal Diesel engine has very little overload capacity, any building up of torque will cause a reduction in its speed, and therefore, its output is automatically limited. The inherent characteristics of the d-c. motors and generators are such that they will carry the increase in torque without the least danger of becoming unstable, and hence no special precautions need be taken. There is always a stable couple between the generators and the motors.

Bridge Control. Since the engines operate at practically constant speed, and in the same direction at all times, and are under the control of a constant speed governor, they require no attention during maneuvering. This combined with the absence of such factors as steam pressure, boiler fires, priming, etc., make control of the propeller machinery from a remote location entirely feasible. In other words, the ship is controlled just as easily from the bridge as from the engine room. The importance of such performance is obvious in the case of ships, requiring very accurate maneuvering in restricted places, as it eliminates both delay in response to signals, and risk as a consequence of mistaken signals. The Diesel-electric system is, in fact, the only system of ship propulsion that affords bridge control without resorting to complications which are questionable at best.

Torque at Low Speed. In regard to low speed torque, the

Diesel-electric system here described is very similar to the turbine, in that it is capable of developing large overload

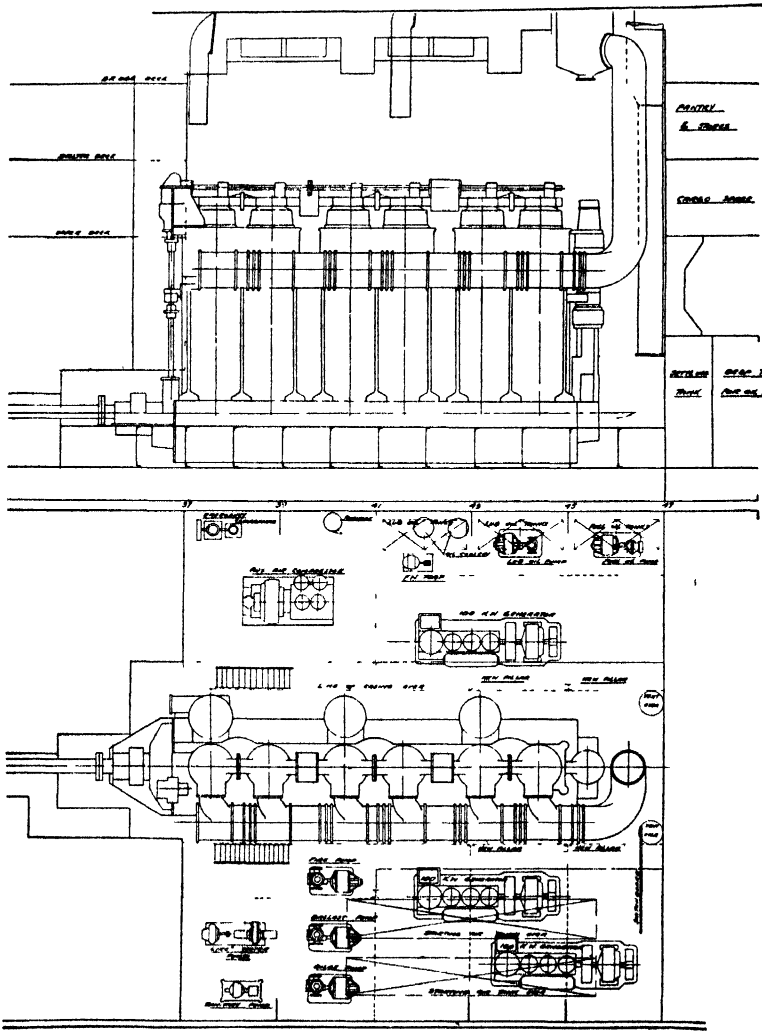


Fig. 286

torque at reduced speed: Theoretically, the torque at the motor shaft may be increased in inverse ratio to the speed without overloading the engines. However, safe commutation

of the electrical machines of ordinary design places a practical limit on the torque that can be developed under these conditions. For the purpose of ship drive, however, the ordinary direct-current machine will develop sufficient torque to meet any emergency.

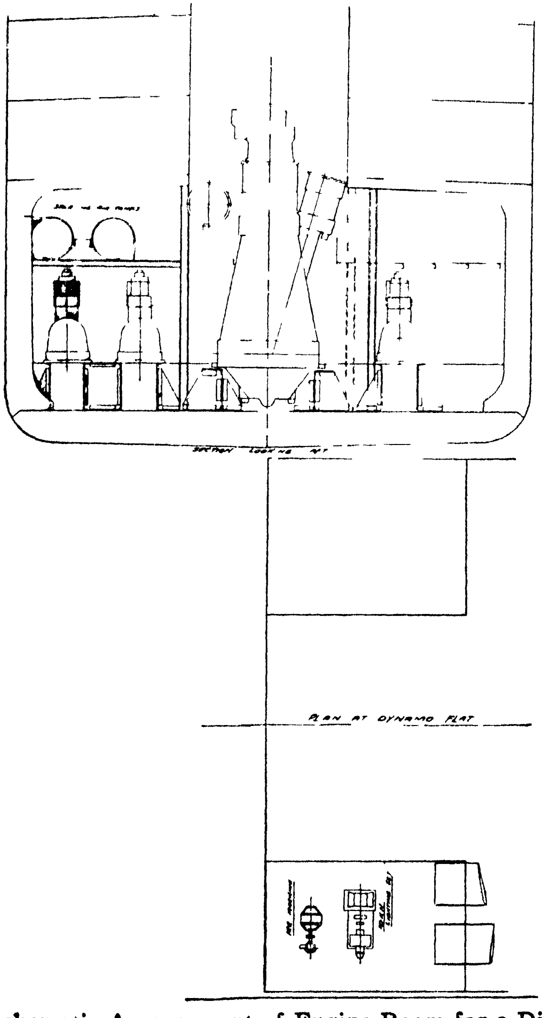


Fig. 286.—Schematic Arrangement of Engine Room for a Direct Diesel Drive for a 9,400 D.W.T. Shipping Board Vessel, Design 1015, with Dimensions L.O.A. of 402 ft. 6 in., Beam, 53 ft.; Draft, 26 ft. 6 in. The Drawing Suggests the Layout for the Diesel-Electric Auxiliaries

The advantages of Diesel-electric propulsion as compared with any form of steam drive are very pronounced.

Fuel Economy. The most important advantage is that resulting from the difference of fuel economy. A properly designed Diesel-electric propulsive equipment requires about 0.55 lbs. of oil per shaft horsepower hour for all purposes, whereas the average high-grade steam installation requires about twice this amount, or more. The saving in fuel is of two-fold importance. First, the actual difference in fuel cost, and second, the additional cargo that can be carried due to the decreased weight of fuel, or the greater cruising radius for a given amount of fuel oil.

As compared with a direct connected twin screw Diesel drive, the single screw Diesel-electric is about on a parity in regard to fuel consumption, but considerably lighter in weight.

Weight. A proper Diesel-electric propulsive equipment being lighter than the geared turbine equipment of high grade performance, enables additional cargo to be carried in the amount of the weight difference. The importance of this feature is apparent as it affects the ultimate earning ability of the ship.

The direct-drive Diesel has demonstrated its superior economy in the earning power of the ship as compared with the steam drive. The result was accomplished in spite of the excess machinery weight of the former as compared with the latter. Similarly, as the Diesel-electric is considerably lighter than the direct-drive Diesel, the Diesel-electric will show superior earning power as compared with the direct-drive Diesel, particularly on long runs.

Reliability and Reserve Power. By providing a number of small generating units, the reliability of the Diesel-electric drive is superior to that of the direct-connected Diesel drive, both from the standpoint of individual engines, and the drive collectively, in the case of casualty. This superiority of the Diesel-electric also obtains when compared with the steam drive.

Reserve power in case of casualty to any of the generating units is important. Having a number of generating units,

the reliability in case of casualty is infinitely greater than in the case of a single-screw steamship. Although with a cross-

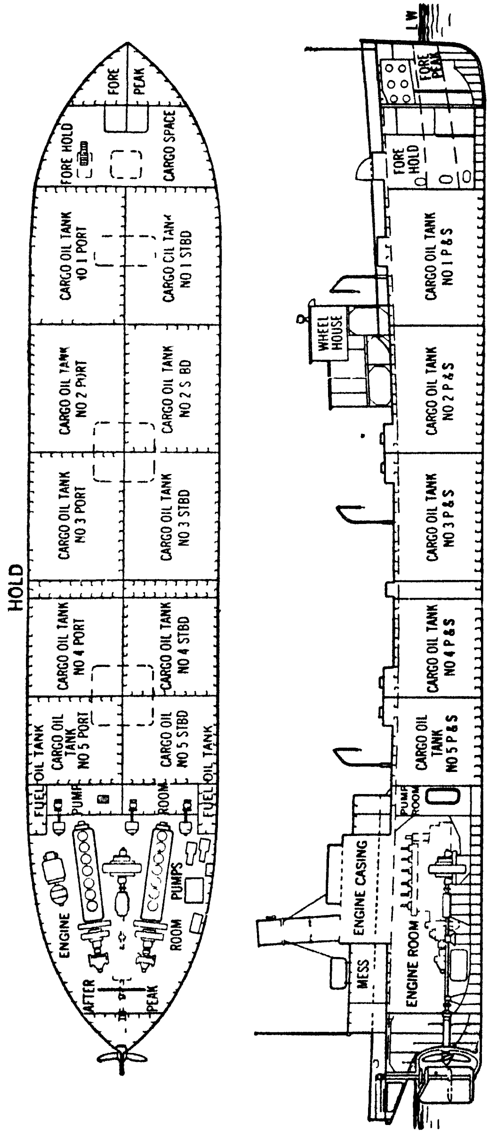


Fig. 287.—Diesel-Electric Tanker, "J. H. Senior," of the Standard Oil Company of New Jersey

compound geared-turbine, the failure of one element would still enable operation at about 50% power from the remaining

element, it would be at a sacrifice of considerable speed and economy. A similar analysis applies in the case of the twin-screw direct-connected Diesel. To provide a motor of small diameter it is customary to use what is known as a double unit motor which, incidentally, results in greater reserve power flexibility. The double unit motor essentially consists of two electrically independent motors mounted on the same shaft. With such an arrangement, the reliability and reserve power of the Diesel-electric in case of casualty, is infinitely superior to that of the single turbine and single direct-drive Diesel single-screw ship.

With the Ward-Leonard system, using the series arrangement of machines, more reserve power is available in case of casualty to a prime mover than is the case with any other system of ship propulsion. Taking for example, a 3-generator, single-screw arrangement, the failure of one generating unit would enable 88% speed to be obtained with the remaining two sets, and in the case of the failure of two generating sets, the single remaining set would furnish sufficient power to propel the ship at 70% speed. This analysis is based on the driving power varying as the cube of the speed. This system is in fact the only system that permits full power to be derived from remaining units without overloading them, or increasing the original size and weight.

To obtain the full capacity of the remaining units under conditions stated in the foregoing paragraph, it is merely necessary to set each generator for its full rated voltage, and then to decrease the motor field current until full load armature current flows through the system. In the example cited, full load current with two generators in operation would supply two-thirds of the total power, and with one generator in operation, would supply one-third full power. It is necessary to weaken the motor field to obtain the required speed for the remaining power, as otherwise the motor would operate at a speed directly proportional to the total remaining generator voltage. If the motor is of the single armature type, its field flux would be reduced to 76% to obtain 88% speed with two generators in operation; and 47½% with one generator in operation. If the motor is of the double unit type with its

armatures normally operated in series, 70% speed can be obtained with one generator, and one motor unit, and in this case this motor field flux is reduced to about 95% of full value.

In the case of a twin or multiple-screw ship having Diesel-electric propulsion, a casualty to a prime mover does not prevent supplying balanced power to all screws. The switching is so arranged that the generators may be connected to any of the motors. This is of further advantage in that the remaining prime movers are operated under normal power conditions and consequently normal efficiency.

Recalling the reliability which was discussed above, it is inconceivable to imagine a reasonable condition of casualty in the case of Diesel-electric propulsion that would prevent the ship from reaching port at a reasonable speed.

Simplicity. The characteristics of the engine for Diesel-electric drive are constant speed and reasonably close regulation from no load to full load. Since the engines operate at constant speed and always in one direction, it is unnecessary to point out the elimination of the reversing gear, as well as the air for reversing. The result of the elimination of these two features means a simpler and probably a more reliable engine. It at least reduces the air problems to their simplest terms. Air is used only for the initial start in port. In fact, the plant can be arranged so that only one engine need be started by air and subsequent engines started electrically by utilizing their generators as motors. Furthermore, as an extreme arrangement, starting air may be eliminated entirely by providing a small gasoline or kerosene engine generator set to do the starting of the main engines electrically.

Stand-by Conditions. Since the Diesel engine consumes fuel only when running, a further economy is effected by the elimination of stand-by losses. Also, it is unnecessary to warm up various pieces of machinery, such as boilers, turbines, etc., for a long period prior to getting under way. The Diesel-electric system can be made ready for sailing on short notice.

Conclusion. From the foregoing discussion, it will be obvious that the Diesel-electric system of ship propulsion using series connected d-c. machines operating on the Ward-Leonard principle, is considerably more than a mere electric

coupling or gear. It is a system containing very pronounced features which are of direct advantage to the improved performance of the ship. The fact that the electrical machines constitute a reliable and flexible substitution for gears and magnetic couplings is merely incidental.

Specific Advantages of Diesel-Electric Drive. Having now considered, in a general way, the advantages of Diesel-electric propulsion, specific advantages, as applied to each particular class of ship, will be discussed. This type of propulsion can be applied—and applied economically—to almost every class of vessel from the small pleasure yacht to the largest of the ocean-going craft.

Cargo Vessel. For cargo vessels, in trans-atlantic or coastwise trade, Diesel-electric propulsion can be applied very economically, from an actual dollars and cents point of view. First, consider the saving in machinery weight which results from using Diesel-electric propulsion. It has been authoritatively stated that for vessels in the 2500 H.P. class, a saving in weight of from 100 to 120 pounds per S.H.P. can be effected by substituting Diesel-electric for direct Diesel drive. When steam drive is considered, with the addition of the necessary boilers, etc., the saving becomes even more marked.

This saving in weight of the propelling machinery has two immediate results. First, for the same size vessels, it permits the carrying of additional cargo. Secondly, for the same shaft horsepower and the same cargo it permits a higher speed. Let us take a specific case for example: If in a 9000-ton vessel, the machinery weight can be reduced 250 tons by the substitution of Diesel-electric for some other type of drive, that vessel can either carry 250 tons more cargo, or it can travel with the same amount of cargo, but at a higher speed, because of her lightened condition, than she was able to do before with the same power.

With the Diesel-electric system, as has been explained before, the exciter units can be used to supply the normal auxiliary load. Here, then, weight is again saved for no large auxiliary sets are necessary. While in port one of the main engine-generator sets can be run for auxiliary purposes, if necessary; for the control is so arranged that any of the main

units may be taken off the main propulsion bus and thrown on the auxiliary bus whenever it is desirable, whether at sea or not.

Those who are considering the conversion of a steam vessel to Diesel drive would do well to consider the saving in structural changes on the vessel itself which are effected by install-

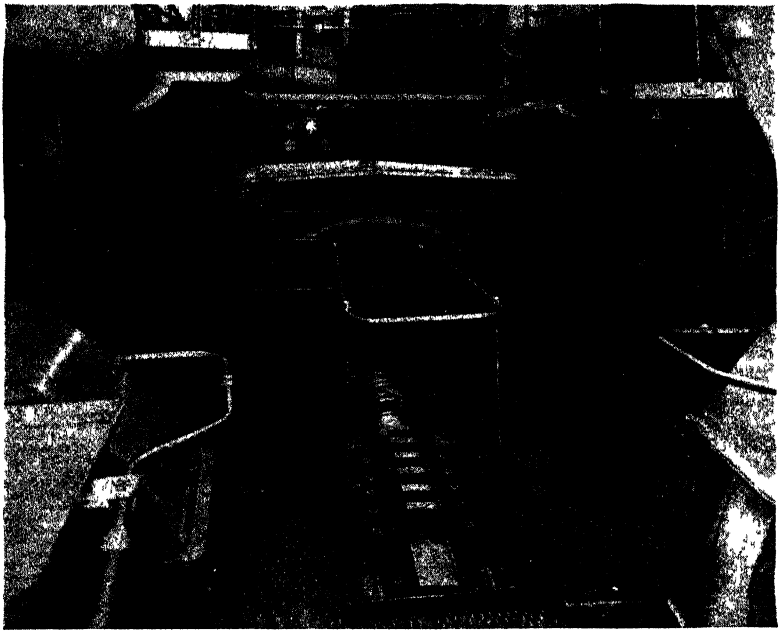


Fig. 288.—Looking Forward in Engine Room of Diesel-Electric Tanker "J. H. Senior"

ing the Diesel-electric system. Most of the present day vessels designed for steam drive are none too strong across the bottom. They are, as a rule, not strong enough to bear the concentrated weight of the large heavy direct-drive Diesel engines available today for horsepowers of from 2000 to 3000. With the direct-drive, then it is, in the great majority of cases, necessary to strongly reinforce the vessel across the bottom before the direct Diesel engine can be installed. With the Diesel-electric, however, no such reinforcement is necessary

because there will be from four to six or more engine-generator units. These will be evenly distributed athwart ship. There will be no large concentration of machinery weight.

Due to the even application of power obtainable with the electric motor, it is not necessary to increase the size of the propeller shaft as is often the case with direct Diesel drive when converting from geared-turbine equipment. This saves money, again, in structural changes.

Due to the fact that one or more of the main generating units can be shut down while the vessel is being operated at reduced speed, the other sets being operated at full power and speed, a very economical set-up is obtained with the Diesel-electric system.

We have already pointed out, in a previous discussion, the reliability and flexibility features which are inherent in the Diesel-electric drive. This feature of reliability cannot be considered too seriously by the shipowner who contemplates conversion. In the Diesel-electric cargo vessel, the possibility of complete breakdown of the propelling machinery is very remote indeed. Consider a vessel say with four engine-generator sets for propulsion. Even if three of these sets should be out of service at one time—a possibility which is very remote—the vessel could still make approximately 64% of her normal speed. This is accomplished by weakening the field of the motor so as to draw full-power from the remaining generator. The propelling motor, for single-screw vessels, is generally made in two units, that is, double armature, so that should anything happen to one unit it can be cut out of the circuit and the ship propelled by means of the remaining armature. This system, then, practically eliminates the possibility of laying up the vessel, or of ever having to tow in the vessel due to breakdown of the propelling machinery. This is, sooner or later, sure to amount to a large saving in money.

If the vessel is equipped with pilot house control, much time can be saved in maneuvering the vessel for the pilot can obtain any speed he wishes, either ahead or astern, by simply moving his control handle in the pilot house. The propelling

motor responds directly to him. No signalling of any kind is necessary, hence no time is lost in transmitting signals and there is no danger of damage resulting from mistaken signals. Since the engines revolve at a constant speed and in the same direction all the time, they do not need the constant attention necessary with other systems and the engine room force may be reduced to a minimum.

Another point which is advantageous, when Diesel-electric drive is concerned, is the fact that with this type of drive the

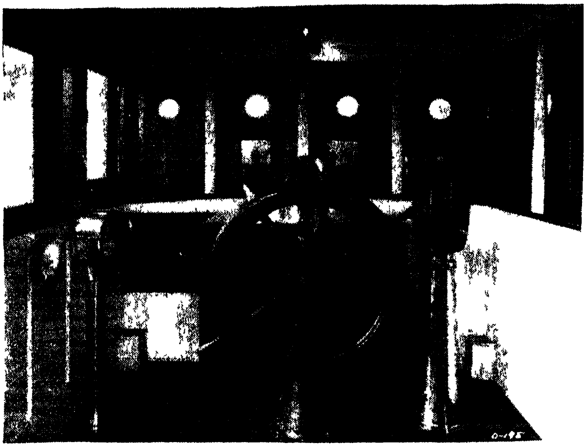


Fig. 289.—Typical Yacht Pilot House, Showing Main Motor Control Pedestal to the Right of the Wheel

center of gravity of the propelling machinery is much lower in the vessel than with any other type of drive. A glance at the layouts on the previous pages show this. The reason for this is, of course, that all of the units are small and they are evenly distributed. With other types of drive, whether using steam or Diesel engines, the center of gravity is sure to be higher, in the one case, on account of the boilers and in the other because of the large engine or engines necessary. The direct result of this lowering of the center of gravity is the stabilizing of the vessel so that she rides better in a sea.

In order to make clear the points just discussed, an economic analysis comparing direct Diesel and Diesel-electric

drive, and including the costs of conversion from steam to Diesel or Diesel-electric, is included below. The drives compared are from 2650 to 2800 S.H.P.

In comparing the two types of drives, the requirements may be enumerated as follows: Reliability, weight, space, installation, first costs, fuel consumption, and operating costs.

Reliability—Reliability should be the first consideration in selecting any propulsive equipment. Compared with single-screw direct Diesel drive, the Diesel-electric drive has the following distinct advantages:

(a) A number of units, provide reserve power in case of casualty to one engine. For example, in a four engine installation, the following ship speeds may be maintained, based on the power varying as the cube of the speed:

4 engines—100% ship speed
3 engines— 91% ship speed
2 engines— 79% ship speed
1 engine — 64% ship speed

The electrical system is such that full power may be obtained from each engine in operation.

(b) The smaller cylinders and absence of oil cooled pistons and reversing gear is decidedly conducive to lower maintenance by virtue of the simplification of the engine and plant.

(c) Smaller and lighter parts greatly facilitate repairs. This feature, together with the flexibility resulting from the use of a number of engines, makes repairs at sea and routine inspection and overhaul possible while the vessel is under way. Also, the facility of correcting minor troubles as soon as they develop, with comparatively negligible loss in ship speed, will reduce maintenance to a minimum and approach 100% plant operation. This also adds to the contentment of the engine room personnel.

(e) Perfect and instant control from the bridge eliminates all possibility of mistaken signals and thus safe-guards the ship in restricted waters.

(f) The compressed air problem is reduced to its simplest terms since the engines operate at constant speed and in one direction at all times regardless of propeller maneuvers.

(g) The addition of electrical apparatus introduces no hazard for the reason that its thorough reliability is established beyond a doubt. The electrical system employed, as previously pointed out, is the simplest possible and easily understood. The motors and generators are simple and require little attention.

Weight. The difference in weight varies with the engines selected in both cases. An analysis of total machinery weights, including deck and engine room machinery, foundations, ladders, gratings, structural work, etc., based on three types of direct-connected engines and three types of Diesel-electric drive engines suitable for delivering 2650 to 2800 shaft horsepower discloses an average weight difference of 116 long tons in favor of the Diesel-electric. In this analysis, the direct-connected engine speeds cover a range of 85 to 105 R.P.M. (Some of the direct-connected engines are heavier than the average direct-connected engine used in this analysis.)

To evaluate the weight saving in the case of the Diesel-electric drive, it is fair to consider that one-half the difference would be used for additional cargo. This factor of one-half allows for the possibility of not being able to obtain the additional cargo on some trips and of carrying cargo on which a lower rate of freight is paid, on other trips. Assuming that the vessel travels 42,000 miles per year and that the freight rate is $5.00 per 1000-ton miles, the additional yearly earning capacity of the Diesel-electric ship over the direct-connected Diesel vessel is

$$\frac{116}{2} \times \frac{42000}{1000} \times \$5.00 = \$12{,}200.$$

Conversely, if it is desired to consider a given cargo in both cases, slightly less power would be required for a given ship speed, in the case of the Diesel-electric vessel.

The advantage of being able to handle 116 tons additional peak cargo at certain times will be obvious to owners and operators.

Space. In comparing space requirements, the differences are found to vary considerably with the engines used. Owing to the lesser head-room required in the case of the Diesel-

electric drive, certain overhead space is available for cargo without changing the present arrangement of flats. This is plainly shown in the engine room layouts for the Diesel-electric drive. These layouts show that all the machinery may be conveniently located in the space normally allowed for the steam propelling machinery.

Installation. The average total installation costs including placing and aligning of machinery, wiring, piping, foundations and structural changes are in favor of the Diesel-electric by approximately 15%.

The Diesel-electric drive allows of better distribution of weight as the engines are entirely independent from the propeller shaft and may be located so as to best suit the space and weight distribution requirements with minimum changes to the present structures. Also, another important advantage of the Diesel-electric drive is the fact that no propeller shaft changes are required at present propeller speeds and 2800-shaft horsepower, because of the constant torque exerted by the motor or, conversely, no reduction in power over the present power at present R.P.M. is necessary with the Diesel-electric. However, with the direct-connected Diesel, due to its varying or pulsating torque, either the present shafting will have to be replaced by a larger one, or the power at present R.P.M. must be reduced, or the speed at 2800-shaft horsepower, increased over the present value in order to come within the rules of the American Bureau of Shipping. Should the power or speed be changed it is quite likely that new propellers would have to be fitted, resulting in additional costs.

First Costs. An analysis of the total costs of installation, including total machinery installation (deck and engine room), foundations, etc., based on the average of acceptable direct-connected engines as compared with perfectly suitable and reliable Diesel-electric drive shows that the latter costs no more, and in some cases less, than the direct-drive, in instances where the present shafting does not have to be changed. Where shafts must be replaced because of direct-drive, there is an additional cost for the direct-drive of $30,000 or more. Where speeds are increased so as to bring the shaft requirements within the limits of the rules of the American Bureau

of Shipping at 2800-shaft horsepower, with direct Diesel drive, new propellers at a cost of $8,000 to $10,000 are required. While it is recognized that double-acting two-cycle engines will require shafting only slightly in excess of that for the constant torque electric drive, it is nevertheless true that this type of engine cannot be considered as an established development at the present writing. Furthermore, any advantages incident to this engine as a direct-drive appliance will apply in like proportion to electric drive engines. The foregoing relates to average conditions: for local business reasons, isolated examples may be at variance with this analysis.

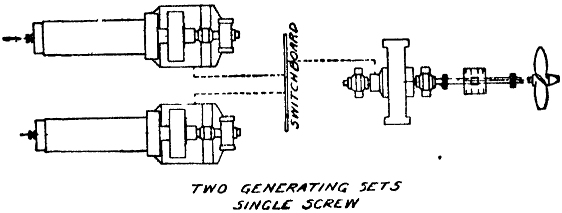


Fig. 290.—Two Generating Sets, Single Screw

However, such cases cannot be considered as representative of general market conditions.

Fuel Consumption. (a) At sea. In regard to the fuel consumption it is admitted that the fuel cost will be less for the direct-drive than for the Diesel-electric. However, a consideration of the actual amount of this difference will show that it is usually a greatly overstated one.

Due to the introduction of the electrical units, the Diesel-electric drive will require approximately 0.485 lbs. of oil per S. H.P. hour as against 0.39 lbs. per S. H.P. per hour for the direct-drive.

A deduction of from four to five per cent must be made from this difference due to:

Constant torque of the Diesel-electric as compared with the cyclic variation in torque of the direct-drive.

Average difference in propeller speed—the lower speed of the electric drive giving higher efficiency.

Percentage of operation at reduced power (Diesel-electric

can be operated at full-load unit fuel rate, at reduced speed, whereas the economy of the direct-drive decreases with reduced power and speed).

These are in favor of the electric drive.

(b) In port. In port, the average vessel requires about 300 K.W. in auxiliary power when deck winches and other auxiliaries are in use, and about 50 K.W. without winches.

With the Diesel-electric drive, provision can be made to supply the auxiliary power from one of the main engines. While this means a somewhat higher fuel consumption on account of using an oversize unit for auxiliary power, it eliminates the first cost and carrying charges on the 3-100 K.W. auxiliary engine-generating sets usually used with direct-drive.

The Diesel-electric ship usually has about a 50 K.W. auxiliary set for use in port when the winches are idle. This in turn would show a fuel saving over the larger capacity 100 K.W. set which would be used at reduced load on the direct-drive ship.

In addition to the above, at sea the auxiliaries would all be driven by power from the main generators or exciters, whereas the direct-drive would require one of the auxiliary sets in operation. This would show a decided saving in favor of the Diesel-electric.

For tankers, where even larger auxiliary capacity is required to operate cargo oil pumps, an even more favorable factor in favor of Diesel-electric drive is found.

Taking all the above into consideration, it is found that while the overall fuel consumption of the direct-drive is somewhat less than the Diesel-electric, the net difference per year is small, and far less in amount than the additional income derived from the extra cargo tonnage capacity available with Diesel-electric drive.

Operating Costs. A study indicates that the personnel should be the same for both direct Diesel and Diesel-electric drives, for single-screw ships. If twin-screws were required on account of the limited direct Diesel engine capacity, the advantage would be with the Diesel-electric drive in the smaller number of men required.

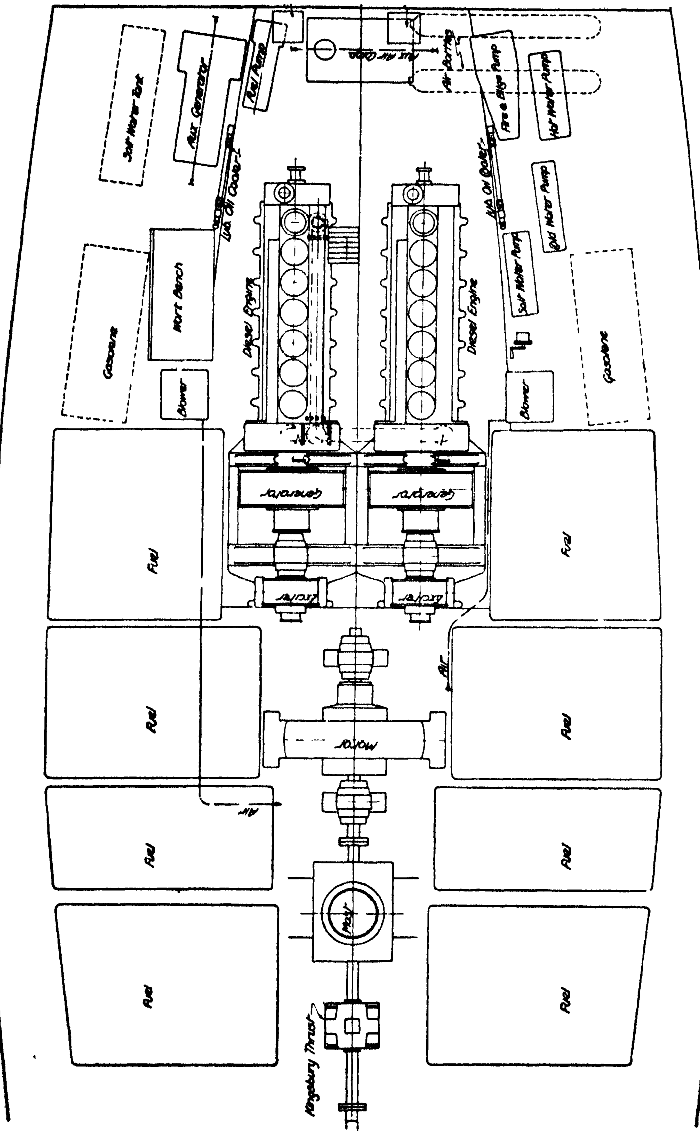


Fig. 291.—Typical Diesel-Electric Engine Room Layout for Yacht

Taking all operating costs into account the total net saving is found to be in favor of the Diesel-electric drive by amounts ranging from four to ten thousand dollars, depending upon the service, auxiliary power required, etc. Capitalizing this at 15%, the owner could afford to pay from 26,000 to 66,000 dollars more for Diesel-electric than for straight Diesel drive. To this amount must be added the difference in installation costs, foundations, shaft sizes, structural differences, etc., when it will be found that in very many cases, it would be most profitable to install the Diesel-electric drive. This would be particularly true in conversion jobs.

And the owner must never lose sight of the advantages of Diesel-electric drive which cannot always be evaluated upon a strictly monetary value, the many additional factors of safety, the reliability, the better working conditions involved and the flexibility and reserve power available.

The ultimate purpose of a merchant ship is to earn money. From this standpoint the factors of reliability, economy, weight, space, cost, operation and maintenance are involved in the propulsive equipment. Therefore, the type of drive which is best suited is the one which excels in all or the most important of these factors. A careful consideration of the Diesel-electric system will prove its superiority for the cargo vessel.

Oil Tanker. Perhaps some of the best examples of Diesel-electric drive are to be found in the modern oil tankers of today. With this class of vessel we not only secure, by the Diesel-electric drive, absolute control of the vessel from the pilot house, resulting in the saving of much time in docking and maneuvering, but we have an unlimited supply of auxiliary power at our command for operating the cargo pumps and the auxiliary apparatus. Furthermore, it is entirely practical, and it is usually so arranged, that both propulsion of the tanker and cargo pumping may be carried on at the same time if necessary. Since d-c. current is used, necessitating the use of a commutating motor, volatile oils are pumped in safety by separating the pumps from the pump motors by a bulkhead, the shafts passing through stuffing boxes. This arrangement has the additional advantage of

placing the large cargo pump motors in the engine room where they may be easily inspected by the chief engineer of the ship. The control for the cargo pumps may be placed anywhere that is convenient. At times it is convenient simply to operate the cargo pump motors by means of push buttons located at the discharge end of the pump line, a meter calibrated in barrels per hour (for example) being located beside the push button. This type of control is accomplished by having a motor-operated-face-plate contactor controller in the engine room, one for each cargo pump. The push button on deck will start

Fig. 292.—Propulsion Motor on Penn. R.R. Tug

the motor on this contactor panel and this motor will continue to operate, thus increasing the speed of the cargo pumps, so long as the push button is held down. In this manner, any pumping speed within the limits of the pump may be obtained. From what has been said before the reliability and flexibility of the Diesel-electric propulsion system for this class of vessel may be readily understood. This flexibility and reliability coupled with the additional advantages stated above, make it an ideal system of propulsion for this class of ship.

Yachts. With the yacht, the conditions are quite different from those of the tanker, in so far as space requirements for the propulsion machinery are concerned. With the tanker we have broad heavy lines, with the yacht, the lines are fine and the engine room space restricted. With both classes of ship, however, we need flexibility and ease of control, and Diesel-

electric easily fulfills these requirements. Due to the fact that the generating sets may be placed wherever necessary to fit in the best, and that the driving motor may be placed as far astern as possible, this system perhaps lends itself better to this class of vessel than does any other propulsion system. The propelling machinery takes up a minimum amount of space. Furthermore, the pilot house control gives the master of the yacht absolute control without any signalling to the engine room. On one of the recent yacht installations of this type of drive, it is said that the control is so sensitive that it is possible to move the vessel ahead or astern a foot or two only, if desired.

Another factor entering into the advantages of the Diesel-electric system for yachts—as well as for other types of vessels—is that the weight distribution is very much better with the Diesel-electric than with any other drive. For example, a typical yacht installation would consist of, from bow to stern; first, the anchors and chain, the ice and stores, then her Diesel engine and generators, then her propulsion motor and finally the fuel and water tanks. This makes a remarkably well balanced distribution of weights, longitudinally and, due to the low center of gravity of the various parts, the transverse stability also favors the Diesel-electric. All of this tends to improve the yacht's action in a sea.

The yacht owner desires a drive which, first of all, can be operated by him with the minimum amount of inconvenience and care. A glance at the cut will show how simple the control is. The speed of the main propulsion motor is controlled entirely by means of the small pedestal to the right of the wheel. It is not necessary to signal to the engine room, for the motor responds directly to the pilot house control. The meters located on the top of the pedestal show the propeller speed—ahead or astern—and the motor amperes and volts. Two small indicating lamps on the panel show whether the overload contactor is open or closed. Thus the master of the yacht can tell at a glance just what power and propeller speed he is getting—and he does not have to leave the wheel house. His operating force can be cut to a minimum, for the responsibility of maneuvering the engine or engines is lifted from the

engineer. All the engineer must do is keep the engine turning over—the governor on the engine will keep it turning at a constant speed regardless of the load thrown on it by the generator. The engine need never be reversed. It is not even necessary for the engineer to reset the overload relay should it trip out due to a direct short-circuit or some other unusual condition, for the pilot can reset this from the pilot house by simply moving his control handle back toward zero. When the generator voltage and the motor voltage balance each other, the contactor will automatically reset.

An important advantage of the Diesel-electric drive for this class of craft is that it is the cleanest system of propul-

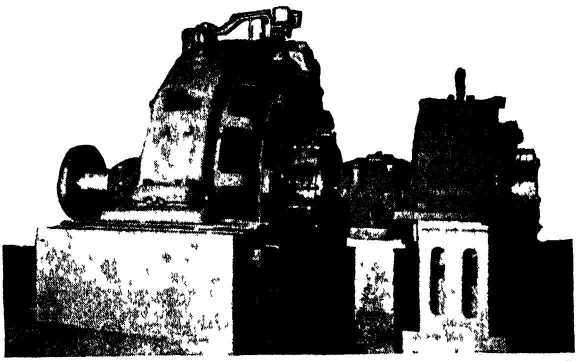


Fig. 293.—Generator and Exciter on Penn. R.R. Tug No. 16

sion yet developed. The electrical machinery makes no dirt, fumes, or smoke whatever, and the Diesel engines, operating as they do at their highest efficiency practically all of the time, make even less smoke than do direct-drive Diesels. There are fewer parts on the Diesels for electric drive than on those for direct-drive, due to the absence of the reversing mechanism.

The Diesel-electric yacht is not only clean, but it may be refuelled quickly and easily and with less confusion and disorder than can other yachts which are not Diesel driven.

There have been a number of installations of Diesel-electric equipment for yachts in the past few years, and, when due consideration is given to the various advantages outlined above, the system is sure to become more and more popular.

Tugboats. The harbor tugboat furnishes one of the most ideal cases, as well as one of the most justifiable applications of Diesel-electric propulsion. A harbor tug must have several inherent operating characteristics, peculiar to its own class of work, all of which are realized in the application of Diesel-electric drive to an extent so far impossible in any other type of drive. These characteristics are as follows:

I. The maximum possible power must be available at all times, whether for (a) light running, (b) medium towing, or (c) working on a condition of 100% propeller slip, that is towing or pushing on a ship, barge, or fleet of barges which are stationary with respect to the water.

II. Maximum maneuvering ability and reliability.

III. Economy of operation.

The above enumerated points are, as before mentioned, the vital requirements of the harbor tugboat. The following is an explanation of the superiority of electric drive on these requirements, over the steam tug and the direct-connected Diesel tug.

I. Consider the character of power application to a towboat propeller. On a free moving ship, between practical limits of speed, the power equals approximately the cube of the speed times a constant. This constant is a function of the displacement and shape of the hull. For this condition of free running, there is a comparatively definite resistance constant, and there is a definite power required for a definite wheel speed. This condition does not exist for tugboats, since the resistance to speed through the water varies from a minimum at the light running condition to a maximum for the maximum tow, or while getting a tow under way.

Consider two tugs with identical hulls, propellers and power, one a straight Diesel or a steam tug and the other a Diesel-electric tug. A steam engine is capable of delivering a maximum torque, corresponding to full steam pressure and maximum cut-off. The possible power from the steam engine is then directly proportional to the R.P.M. of the engine. Since the maximum wheel R.P.M. is attainable only at the light running condition, the maximum horsepower is available only at this condition. From the light running condition, consider

the effect of increasing the resistance of the steam tug as it moves through the water by adding towline resistance. To maintain the same speed, the increased resistance will demand an increased torque and an increased power. Since as stated above, the engine is working at maximum torque, the R.P.M. must decrease until the torque demand is equal to the maximum engine torque. This decrease in R.P.M. causes a decrease in power in direct proportion to the decrease in revolutions. That is, if at the maximum light running conditions, the engine delivers 1000 H.P. at 100 R.P.M., and if the addition of a certain tow increases the resistance to the point where the engine at maximum torque will turn the screw at only 80 R.P.M., the horsepower available for towing at this R.P.M. is only 800 H.P.

A direct-connected Diesel engine represents exactly the same power speed characteristics as the steam engine when applied to a tugboat, and the same analogy applies to both. This is readily seen when it is remembered that the power of both the steam and Diesel engines are calculated from the "PLAN" formula. In this formula, there are generally two variables, "N" the R.P.M., and "p" the mean effective pressure, but the latter, for the purpose of this comparison, is given its maximum value and is therefore a constant. This leaves only the variable "N" which determines the fact that the power available from either the steam or Diesel engine is directly proportional to the R.P.M.

The Diesel-electric tugboat presents a radically different application of the power to the screw. The engines run at a constant speed and are, therefore, capable at all times of delivering maximum power. The electrical link between the engine and the propeller provides a means for delivering the maximum output of the system to the propeller throughout the working range of speed; that is, from the light running condition to the condition of maximum tow, or 100% propeller slip. This is made possible by the fact that the motors for this application are designed for continuous running at maximum power for the most severe working condition to which the tug can be put.

Referring to the curves on Figure 294, curve "S" repre-

sents the power curve of the steam engine, curve "A" represents the power required to drive the tug at the light running conditions. Curves "B" and "C" represent the speed and power of the screw of the towboat when towing, curve "B" representing a light tow and curve "C" a heavy tow, or during the docking of a heavy ship. The intersection of the engine curve "S" and the propeller curves "A," "B" and "C" determine the operating characteristics for the various towing conditions, and it will be seen that the power available from the engine decreases as the demand increases.

Consider curve "E," the power available by the Diesel-electric system. As stated above, the power available is constant throughout the working range of the tug. This is shown graphically by the fact that the power curve "E" cuts the light running, light tow and maximum tow curves "A," "B" and "C" at the maximum, or 100% power value. Compare this with curve "S" and note that the electric tug has a decided advantage, particularly at the lower full-power speeds where the most important work of the tug occurs.

The curves of Fig. 294 are for purposes of illustration of this text only, and should not be used for power calculations.

II. The harbor tug imposes the severest maneuvering condition to which any marine propulsion system can be subjected. Maneuvering is, at times, almost continuous. The speed and certainty of maneuvering is a measure of a harbor tug's ability to perform its work without doing damage. The most important maneuver is from full-power ahead to full astern, or from full astern to full ahead. A consideration of the operations necessary to perform this maneuver discloses the facts that:

(a) The steam engine requires three operations. (1) Close throttle, (2) reverse valve gear, (3) open throttle.

(b) The direct-connected Diesel engine requires four operations. (1) Close throttle, (2) reverse valve gear, (3) apply starting air, (4) open throttle.

(c) The Diesel-electric system requires one operation; that of reversing the main generator fields.

The speed of maneuvering, of course, depends upon the number of operations necessary to bring about that maneuver.

The above comparison can leave no doubt of the superiority of electric drive in this respect. The certainty of maneuvering depends upon two things—the mechanism itself, and the human element.

In a comparison between the direct-connected Diesel and the Diesel-electric systems, it is a conceded fact that more reliable performance in maneuvering is possible with the latter system on account of the fact that the engines for all conditions, run at a constant speed, all maneuvering being done electrically. The direct-connected engine requires the use of

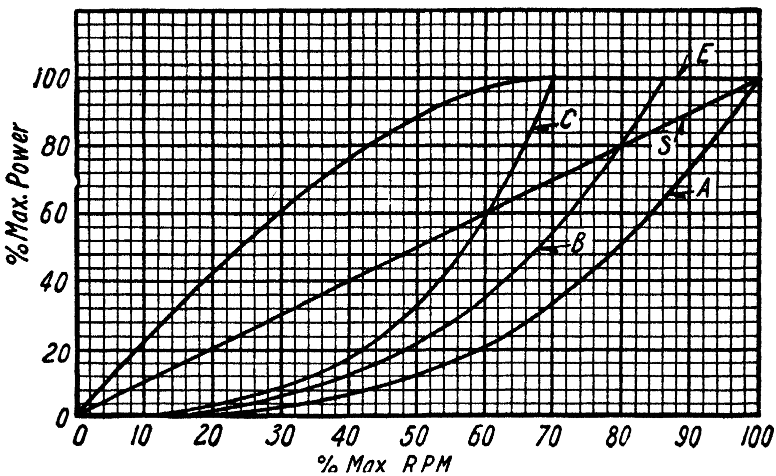


Fig. 294

starting air for every start of the engine, and the manipulation of the cam reversing mechanism for every reversal. One failure of a tugboat engine to start may mean thousands of dollars in damage. The engine builders themselves admit that they would rather apply the Diesel engines to the electric system for this application than to supply a direct-connected unit. There are two fundamental reasons for this: first, the continual use of starting air is detrimental to the engine; and second, a failure to reverse is a direct reflection against the engine.

The fore and aft compound steam engine used in the majority of harbor tugs is ordinarily considered reliable as

to maneuvering qualities, but occasionally sticks on dead center. Serious damage has resulted from this cause.

The reliability of the electric drive can best be illustrated by an example furnished by the steel industry. The reversing rolling mills were all originally operated by large reversible steam engines. Rolling mill service is much more severe than any marine application, being a continual process of reversing. In order to prevent the expensive delays due to breakage and repairs on the steam engines, many of the steel plants have discarded the steam drive and installed direct-connected reversing motors. The electric drive has met the issue in every Westinghouse installation. In this connection, it is to be noted that the main motors and generators supplied for Diesel-electric marine drive are of the steel mill type, with the addition of special marine insulation and non-corrodible brush rigging. It is often remarked that the Diesel-electric system is as simple and easy to control as an electric street car. This idea should be discouraged, because the street car control is a fairly complicated system when compared with the Ward-Leonard system, as used in marine work. For instance, the controller on a street car handles the main current of the motors throughout the entire range of speed control, while with the Diesel-electric system, the main circuits are not changed, all maneuvering being accomplished by controlling the magnitude and direction of generator shunt-fields, the maximum excitation being only 1 to 1½% of the main power.

The advantage of electric drive over the steam engine and direct-connected drive in maneuvering, from a power application standpoint, is even more pronounced than its advantage while handling heavy tows.

Referring to Figure 295, Line "A" represents the maximum torque available with the steam or direct-connected Diesel engine which is a constant for maximum conditions, irrespective of the speed. Curve "B" represents the torque available with the electric drive. The torque with the electric drive varies inversely as the speed of the screw. From the above, it is easily seen that with electric drive, at least double the maximum torque of the steam engine or direct-connected Diesel engine, is available at the propeller at half normal free

running speed. This is particularly desirable when a sudden reversal is required from either full ahead or full astern, and can only be obtained practically, to date, by the electric drive.

The refinement of the electric control cannot be approached by any other practical system yet proposed. Any required speed range from stop to full ahead or astern may be had. The practical value of this feature consists in being able to take a gradual strain on a hawser instead of applying a sudden force on account of the hawser having to stop the momen-

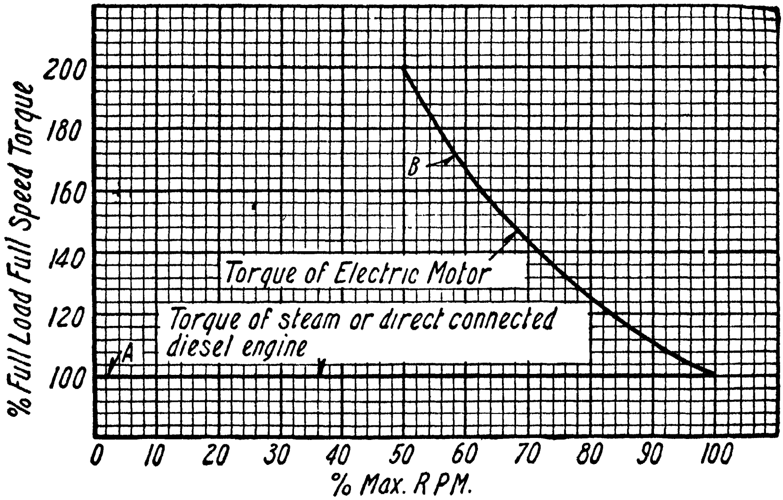


Fig. 295

tum of the tugboat. The class of steam tugs comparable to the Diesel-electric *P. R. R. No. 16* each uses approximately $1,000 worth of towing hawsers per year. It has been estimated that the electric tug will save one-half of this.

The human element is, of course, most dependable at the simplest operations, especially if the operations come in rapid succession. In this respect, the electrical apparatus reduces the chance of failure to an absolute minimum. The electrical drive not only reduces chance of failure due to the personal element to the absolute minimum, on an equal comparison with the other two types of drive with the pilot-house-to-engine room signal system as a basis, but completely eliminates the

personal element in the engine room in the case where the controller is placed directly in the hands of the Captain. It is easier physically and takes less thought and time on the part of the Captain to operate the electric controller than it does to operate the bell pulls on a steam or direct-connected Diesel tugboat. He has the assurance that he can have any condi-

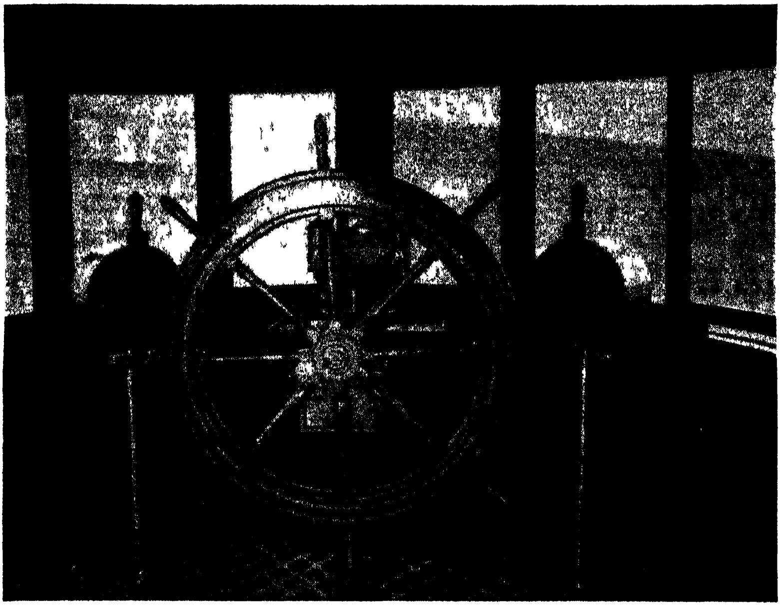


Fig. 296.—View of Pilot House on Penn. R.R. Tug No. 16, Showing the Steering Gear Pedestal in the Center, with the Duplicate Control Pedestals, One on Each Side of the Wheel. These Control Handles Are Tied Together Below Deck with Gearing So That the Operator May Control the Speed and Direction of the Screw from Whichever Side of the Pilot House is the Most Convenient

tion of power he wants without waiting for some one else to interpret his signals and then act upon them. That the bell pull system offers a serious handicap to fast maneuvering is evidenced by the common expression that "the engineer was four bells behind the captain." This lends credence to the belief of one tugboat operator that the electric drive would save 40% of the damages sustained by his firm due to collisions.

III. The operating economy of the Diesel-electric tug over a steam tug lies in reduced fuel bills and reduced crew; and over the direct-connected Diesel tug, in the reduction of the crew.

The following comparison is based on working the tugs with three shifts as in accordance with New York Harbor practice. The figures are fairly close estimates by a tug boat operator on boats of the same size, one being Diesel-electric, one Steam and one straight Diesel drive.

Expenses Per Month. There are several economy features incorporated in the electric drive to which it is as yet impossible to assign a definite value. For instance, much of the work of a harbor tug consists in maneuvering comparatively light barges around wharves and slips. This can be done at half power by shutting down one unit of a two unit drive, and working the remaining unit at its maximum efficiency.

	Steam Tug	Direct-Connected Diesel	Diesel-Electric
Fuel	(a) $1800	(b) $730	(c) $900
Lubricating oil	5	200	200
Grease	9	0	0
Water	40	0	0
3 Ch. Engrs.	570	570	570
6 Firemen	840	0	0
3 Oilers	0	420	0
Total per month	$3264	$1920	$1670

(a) 12 Tons coal per day at $6.00—25 days per month.
(b) 20.3 Gal. oil per hour at 6 cents—25 days per month.
(c) 25 Gal. oil per hour at 6 cents—25 days per month.

The reduction in damage by collision, etc., and the saving in hawsers has already been referred to.

The shut down time for repairs on the steam tugs averages five days per month, and on the Diesel boats is not expected to be over two days per month.

The electric boat will have an inherently higher wheel efficiency during maneuvering over the direct-connected Diesel boat, on account of large low-speed wheel of the former and the small high-speed wheel of the latter.

It has been suggested but not yet proven that with two identical propellers, with the same power input, the wheel driven by the motor will be more efficient than that driven by the compound steam engine on account of even turning moment of the motor and the comparatively uneven turning moment of the steam engine.

It is conceded that the first cost of the Diesel-electric drive

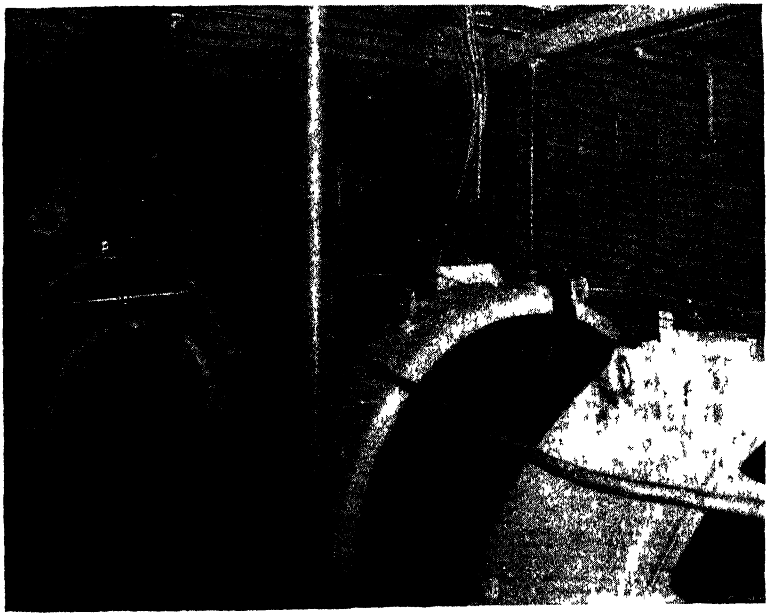


Fig. 297.—View of Engine Room, Penn. R.R. Tug No. 16. The Two Engine Generator Exciter Sets Are in Background and Propulsion Motor is Shown in Lower Right Hand Corner of Illustration

will be more than a steam or direct-connected Diesel drive. However, the management responsible for the operation of the first Diesel-electric tugboat is well satisfied that the extra expenditure is more than justified by the advantages gained.

River Boat. The river towboat is another class of craft on which Diesel-electric control works out to great advantage. There is perhaps no other class of vessel which must do more maneuvering than the towboat plying on inland waterways. They are constantly locking over dams, necessitating quick-

starts and reversals. With the ordinary steam river boat drive, the pilot must be constantly signalling to the engineer. With the Diesel-electric river boat, this signalling is eliminated for the pilot or master, has the direct control of the stern wheel or tunnel screw speed in the wheel house beside the wheel. The whole boat is, in fact, under one man control.

Fig. 298.—Control Pedestal

When coming down stream, one of the engine generator sets may be shut down and the other set operated at full speed and power, this resulting in a much more economical form of drive than could possibly be obtained under any other system.

The feature of reliability, which is inherent in the Diesel-electric system should appeal to every riverman. Instead of relying on one engine, we have two or more engines, any one

of which may be used for propulsion. This latter advantage is particularly to be noted when comparing direct Diesel engine drive with Diesel-electric drive.

Furthermore, the Diesel-electric drive for river boats as well as for all other classes of vessels, provides a refinement

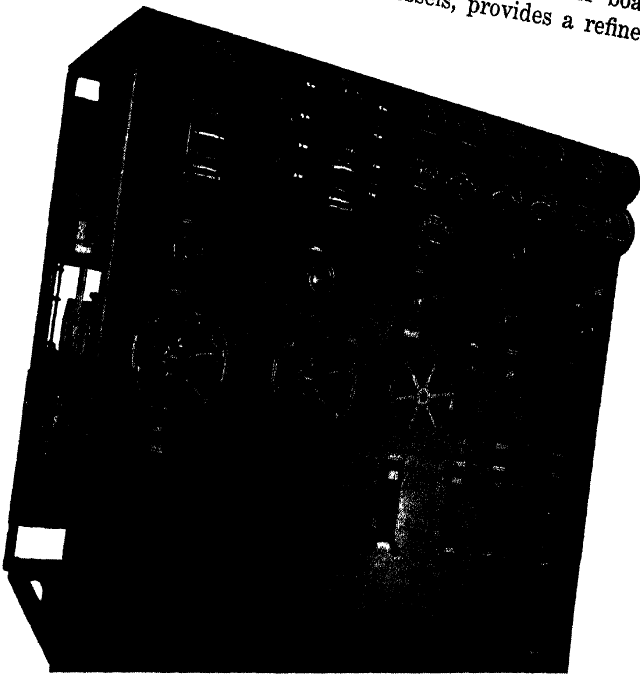


Fig. 299.—Front View of Dead Front Switchboard for Tug Boat

of control not possible with any other drive. Any reasonable number of steps in speed regulation—say from 5 to 25—may be obtained from zero to full speed in either direction. The crew may be reduced to a minimum number for it is not necessary to have a boiler room force, nor engineers to stand constantly over the engines. The engines run constantly in the same direction and at the same speed—they need never be maneuvered.

This type of drive works out particularly well with tunnel-screw river boats, but it can readily be applied to the stern wheel boat. Gears are used in such cases to reduce the motor speed to that suitable for driving the pinions.

Ferry Boat. There is a type of vessel to which the Diesel-electric system lends itself admirably. This is the ferry boat class which will now be discussed.

Propelling machinery for ferry boats may be classified as side paddle wheel and single or double-ended screw type. The application of electric drive has many favorable points in either case, and the electric power is applied with sound mechanical analysis. In the case of screw propellers, the motors are direct-connected. In the case of paddle wheels, the motors are geared as would be the necessary arrangement with any form of drive except the long stroke reciprocating steam engine. The gears may be either single or double-reduction, depending upon the speed of the wheel; however, since paddle wheels of ordinary design are inherently low-speed devices (14 to 20 R.P.M.), the double-reduction gearing is preferable. The low-speed reduction should preferably be of the spur gear type, to permit lateral motion without shocks to the gear teeth in case of side shocks to the wheel.

There are three general methods of driving ferry boats electrically:

1. Diesel-electric, using direct-current.
2. Turbine-electric, using direct-current.
3. Turbine-electric, using alternating-current.

The relative merits of the three systems are, generally, in the order named. Only in those cases where the cost of coal is very low in comparison to oil or where oil itself is very cheap, do either of the latter show to advantage, and incidentally, such cases are very rare. Practically speaking, the fuel consumption of a Diesel engine is a definitely known fact, whereas on the other hand, the fuel consumption of steam installations varies considerably, and is appreciably affected by the human element in attendance. Unless extreme vigilance is exercised in the operation of fires and boilers, and the latter maintained in good condition, test results which are made under ideal conditions are not attained in practice. In

the case of the Diesel-electric installations, the engines are provided with governors which automatically regulate the flow of fuel to that actually required, thus eliminating the human element from this important function.

The difference in fuel consumption, while under way, is further augmented by the fuel consumption during lay-over periods. In intermittent service, such as is the case with ferry boats, the fuel consumption of the steam drives during the lay-over periods is appreciable, while the shutting down of the engines in the case of the Diesel-electric drive eliminates lay-over or "stand-by" losses entirely for the propelling machinery.

While the fuel saving is great, it nevertheless does not represent all the saving possible. Owing to the elimination of the boiler plant, the Diesel-electric plant shows an appreciable saving in propelling machinery personnel and under certain conditions, a saving in maintenance.

Similarly, turbine-electric drive possesses economic advantages when compared with the reciprocating steam drive; however, not to the same degree.

Besides the economic advantages, there are a number of intangible advantages inherent in some of the electric drive methods, such as:

(1) Bridge control.
(2) Maneuvering.
(3) Reserve power in case of a prime mover casualty.
(4) Suitability to double-ended propeller drive.

The Diesel-electric drive can be controlled from the bridge or pilot house very readily, without introducing any complications whatsoever. In fact, duplicate control stations can be placed wherever desired. In the preferable form of Diesel-electric drive, the generator voltage or Ward-Leonard system of control is used with the generators connected in series. As described elsewhere in this issue, this system is the acme of simplicity. Placing the control of the propelling machinery directly in the hands of the pilot greatly improves the rapidity and facility of entering slips. This provision also enhances the safety of entering slips, as the boat can be more uniformly steered and not jeopardized by the possibility of mistaken sig-

nals in the engine room. The importance of bridge control is obvious in the case of any ship maneuvering in restricted waters.

While bridge control might be arranged with either of the two turbine systems, it would involve some complication because of the desirability of simultaneously regulating the fires, particularly with oil-fired boilers. In the case of the turbine-

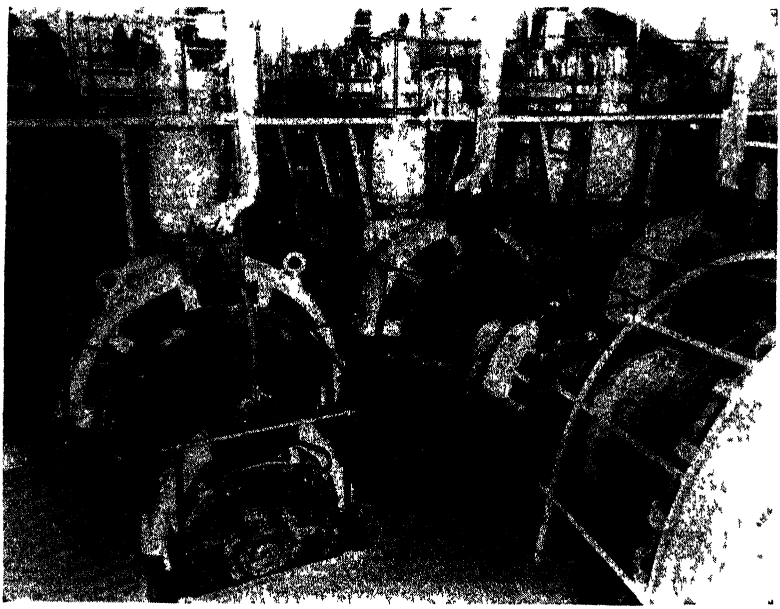


Fig. 300.—Engine Room Layout of the "J. W. Van Dyke." Three Diesel Engine Driven Generators Driving a 2,300 S.H.P. Motor

electric using a-c. machinery, automatic control would be required for the electrical and turbine machinery as an additional complication.

In maneuvering, the electric drive cannot be surpassed. Either the Diesel-electric or the turbine-electric using d-c. machinery possesses a refinement of speed control and reversing facilities beyond that of any other system. Here any number of speed points in either direction can be had, without

detracting in the least from the simplicity. The Diesel engines or turbines run at constant speed regardless of the propeller speed or its direction of rotation. This great simplicity and refinement are accomplished by simple adjustment of the generator field current.

Using two or more generator units provides reserve power in case of casualty to prime movers. With either of the d-c. systems described, full power of the remaining units is available for supplying the propeller. In a 3-unit plant, the vessel can be driven at 88% speed with two units in operation. A-c. systems do not provide reserve power to as great extent.

In the double-ended ferry, marked economy is obtained by running the forward and after propellers at different speeds. The forward propeller is turned over at such speed that it imposes no resistance to the movement of the vessel in the water. All propulsive power is furnished by the after propeller. This speed differential is very easily taken care of by the electric methods. In the d-c. installations, the field of the forward motor is strengthened so as to reduce its speed to the proper value. When once the proper value of field current has been determined, the speed differential between the forward and after motors will remain approximately the same for any speed condition. When driving the ferry in the opposite direction, the other motor is similarly adjusted, and the control arranged to place either weak field or strong field automatically on the motors, depending upon which motor is doing the driving. In the case of a-c. drives, the speed differential must be obtained by different numbers of poles on the motors, finer adjustments necessitating external resistance.

The difference in the speeds of the forward and after propellers depends upon the inherent slip in the propeller when driving and consequently varies with the design. While d-c. installations can accommodate any speed differential, a-c. installations are limited, in that it is only possible to increase the differential established by the different number of motor poles, and that must be done by inserting resistance in the forward motor secondary. The control is arranged to interchange automatically the speed differential when the connections to the motors are reversed.

The General Electric System of Diesel-Electric Drive

The equipment furnished by the General Electric Company for use on Diesel electric driven vessels consists of the generators, both main and auxiliary, the propulsion motors, and the control for the electrical end of the system, and electrical

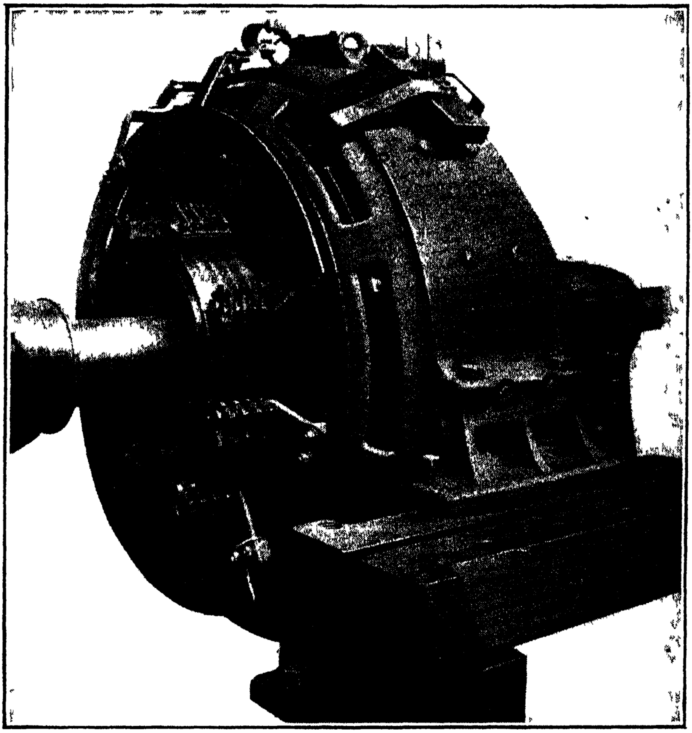


Fig. 301.—General Electric Generator for Direct Connection to Diesel Engines

equipment for auxiliaries. All the ships equipped with Diesel-electric drive up to the present time are using direct current apparatus, and, in view of certain characteristics of direct current, it will probably continue to be preferred to alternating current in the majority of cases. The reasons are simple; direct current permits all speed changes to be accomplished without changing the engine speed, thus necessitating only a

simple control system; when several engines are used it is unnecessary to run the generators in parallel or on separate motors as would be required with alternating current machines. The fact that with direct current generators the prevailing method is to connect them in series adds to the simplicity of the system.

Generators. The main propulsion generators used for Diesel electric drive are type MPC, usually shunt wound, direct current machines, designed for direct connection to the Diesel engine. These are shown in Fig. 301. The magnet frames are made of cast steel split horizontally so that the top half may be readily removed for inspection or repair. The supporting feet are cast integral with the lower half of the frame. The main field poles are made up of punched steel laminations that are riveted together and bolted to the magnet frame in such a manner that they may be removed without removing the armature.

The shunt field coils are wound with pre-treated cotton covered wire on insulated spools. Before assembly, the complete coils are dipped in varnish and then baked to make them oil and moisture resisting. The commutating field coils are wound with copper strips placed on edge, so as to secure the maximum amount of ventilation and radiation surface. The two end turns are insulated so as to give the maximum creepage surface.

The magnetic circuit of the armature is made up of punchings stamped from fully annealed sheet iron and coated with a special enamel that practically eliminates eddy currents. These punchings are firmly keyed to the armature spider. The armature winding is made up of copper bars thoroughly insulated and held firmly in place by means of wooden wedges driven into grooves in the teeth. Binding bands are used on the end windings to prevent the displacement of conductors by centrifugal force.

The commutator is mounted on a cast iron shell, which is keyed to the armature spider, and is of sufficient length to allow a slight staggering of the brush holders. The commutator bars, made of hard drawn copper, are insulated from each other by mica, and are firmly held in place by cast steel clamp-

ing rings so that no displacement can be caused by expansion or contraction.

The brushholders are made of bronze alloy, and are firmly supported by a cast iron yoke which is mounted on the magnet

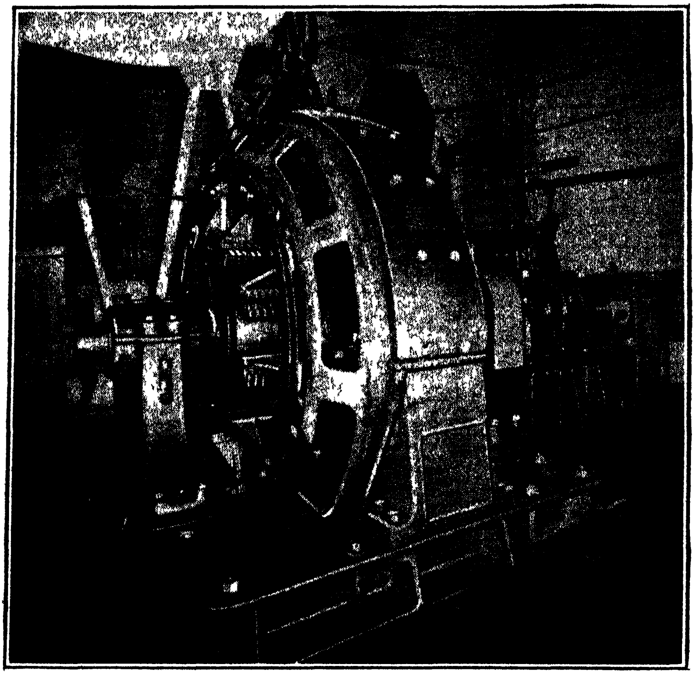


Fig. 302.—750 H.P. Propelling Motor, Built by the General Electric Co. and Installed in the Ferryboat "Golden Gate"

frame. The terminal board is located on top of the machines, and is arranged for overhead wiring.

The auxiliary generators are of the same type and construction as the main generators, with the exception of a few details that are modified because of the difference in physical size and capacity.

Propulsion Motors. The motors used for propulsion are type MPC, usually shunt wound, direct current motors, and are built for direct connection to the propeller shaft. With the exception of some details, they are of the same construction and

general type as the main generators. They may, however, for the purpose of description, be divided into two classes, namely, single armature, as shown in Figs. 302, 303 and 304, and double armature motors, as shown in Fig. 305.

The main points of difference between the two classes are in the bearing arrangements and the method of ventilation. In the single armature motors the bearings are of the ball

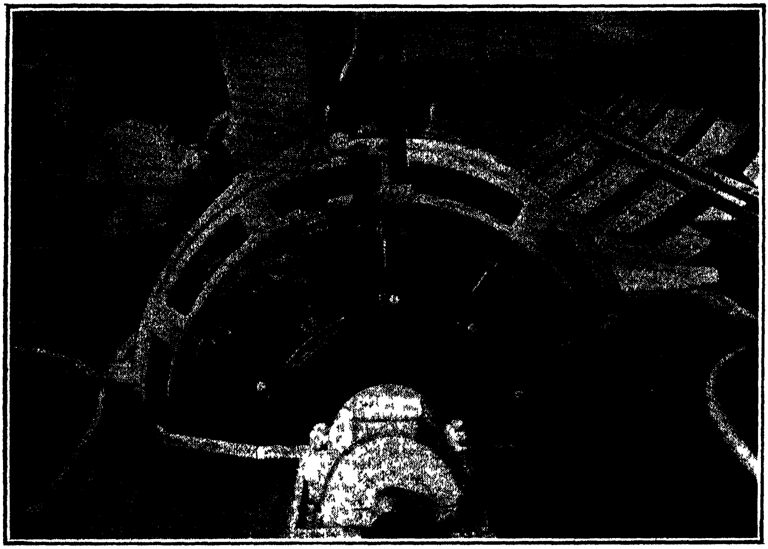


Fig. 303.—370 H.P. Propelling Motor, Built by the General Electric Co. and Installed in the Tug Boat "Van Dyke

seated, self-aligning type, and are split through their horizontal diameter. The bearing shells are lined with babbitt and are provided with disc oiling to insure lubrication. The bearings are so designed that neither a 25 degree roll or list or a 10 degree pitch will cause oil to run out of them and along the shaft. Ventilation is provided by a combination of fan blades on the armature, or an external motor driven fan.

The double armature motors consist in reality of two separate motors mounted on the same base, and having their armatures mounted on the same shaft with the commutators outwards. The shaft is supported by two spherical seated, self-

aligning bearings which are bolted to a common foundation. Ventilation is provided by an external motor driven fan which forces clean air into the chamber formed between the motors by the sheet steel casing which connects the magnet frames. The air passes between the armatures and the fields, and out over the commutators.

Operation Methods of Control. Up to the present two types of control have been developed for use on Diesel-electric ves-

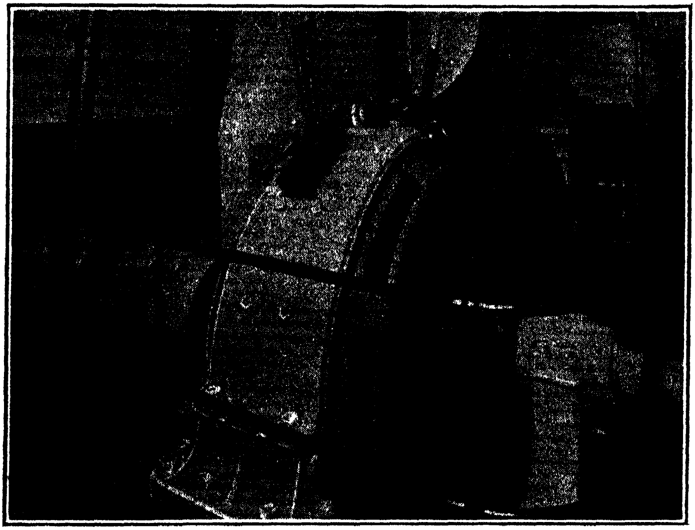


Fig. 304.—Motor Room of the Diesel-Electric Motorship "Twin Ports" Showing Two General Electric Direct Current Propulsion Motors

sels. One of these types is known as rheostatic, or armature control, and the other as voltage control, or the Ward-Leonard system. Examples of both types, with modifications, have been applied successfully in different varieties of ships, the choice in each case having been determined by the ability of the control to meet the requirements of that particular ship. The two systems have very marked individual characteristics which exert considerable influence on the question of their suitability in given cases.

The factors that have most to do in determining the proper variety to use in any case may be briefly outlined as follows:

Ward-Leonard control is much the more flexible of the two, both in the matter of speed control, and of torque. It gives a possible 60 points of speed gradation as against a possible 8 or 12 in the case of rheostatic, and it also permits variation in torque to meet different service contingencies. It does, however, require the use of separate exciters, which are dispensed with in the rheostatic system. Rheostatic control is especially adaptable to ships having a number of electrically operated

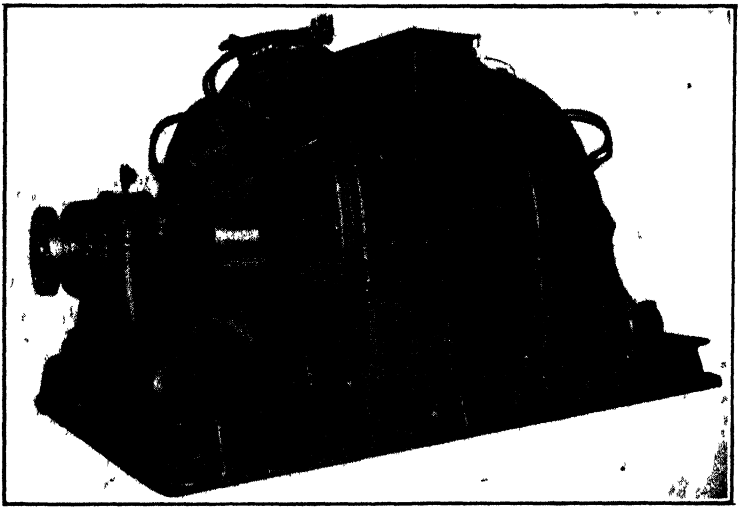


Fig. 305.—600 H.P. General Electric Co. Double Armature Motors

auxiliaries, particularly when these are to be operated simultaneously with the main propulsion motors. This advantage arises from the fact that the main generators are constantly run at full voltage, hence no extra generators are required to insure a steady and adequate supply of power for the auxiliaries.

Rheostatic control is, however, subject to limitations that practically preclude its use on ships of more than about 1,000 H.P. or thereabouts. This limitation may be partly ascribed to the fact that speed variation involves breaking the main power circuit, and partly to the fact that the different gradations in the propeller speed are obtained by the insertion of

resistance in the main circuit. As a consequence, in the first place, as the horsepower increases the necessary contactors become proportionally heavy, bulky and expensive, and the maintenance of proper operation becomes increasingly difficult. Secondly, the size of the resistors necessary becomes unduly

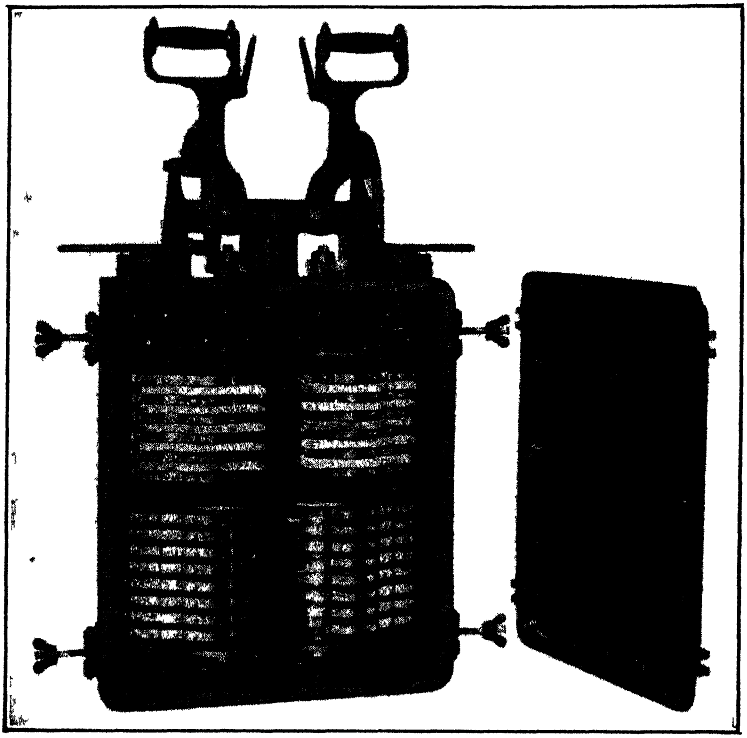


Fig. 306.—General Electric Co. Controller and Switch

great, and large amounts of power are wasted in heat, thereby impairing the efficiency of the whole system.

With Ward-Leonard control on the other hand, the amount of power to be handled has no appreciable effect on the size of the control equipment, and no power is wasted, since only enough is actually generated to take care of the immediate needs of the propulsion motors.

In order that the operation of the two control systems may be clearly understood, a description of a typical example of each is given in some detail.

With rheostatic control, Figs. 306, 307, 308, 309 and 310, speed variation during running periods, and proper acceleration and retardation during starting and stopping, depends upon the insertion of blocks of resistance in the circuit between the generator and the motor armatures. The control

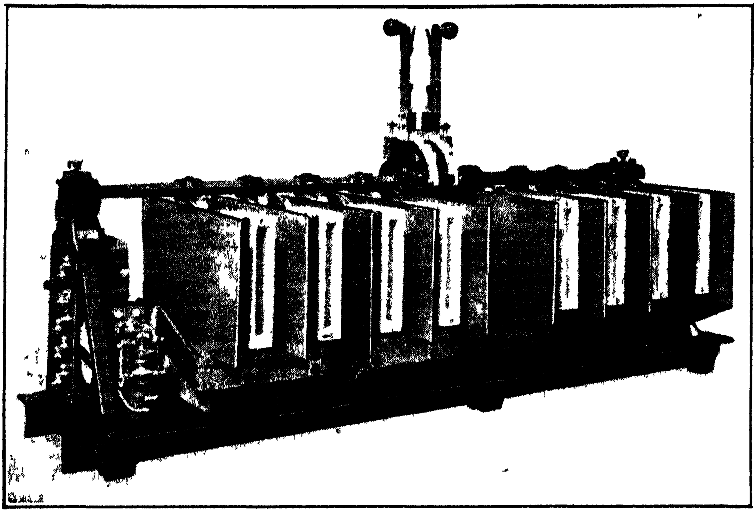


Fig. 307.—Control Group of Rheostatic Control

equipment supplied for this purpose consists essentially of an engine room control panel, a motor room panel, the motor control groups, the starting resistors, the pilot house master controller, if desired an engine room master controller, and the necessary instruments, knife switches for changing the connections between the motors and generators, and protective equipment. While the operation of this equipment is usually electrical, a system of manual operation is provided to operate the control contactors in case of emergency.

The engine room control panel carries the equipment necessary for setting the voltage of the main generators, switches

and circuit breakers for disconnecting the generators from the main bus, rheostats for regulating the fields of the propulsion motors, motor cutout switches for disconnecting them from the line, and switches for transferring the control from the engine room to the pilot house and vice-versa.

The motor control groups contain the contactors for reversing the motor armature, and those for cutting in and out the

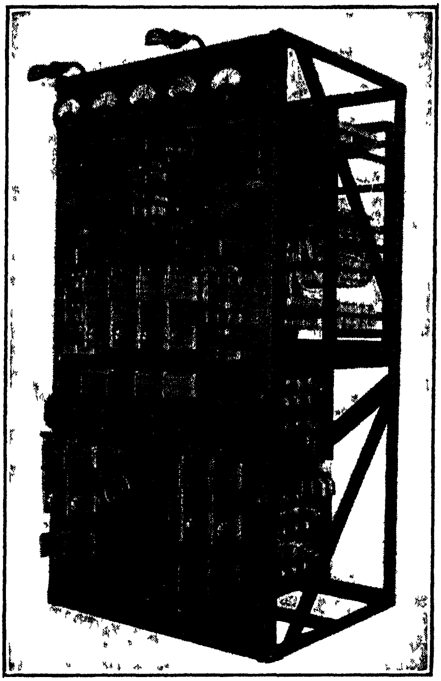


Fig. 308.—Panel Used with Ward-Leonard Control

blocks of starting resistors, two levers for operating the contactors manually in emergency, and an overload relay to trip out the armature reversing contactors when the armature current becomes excessive. The starting resistors are also located in the engine room, and are so designed that it is possible to run on any resistor point continuously to obtain different speeds.

The operation of the propulsion motors is controlled through the motor control groups by either the pilot house or engine room master controller, Fig. 306. In detail, the operation is somewhat as follows: When the master controller is turned to the first point from the "off" position, one

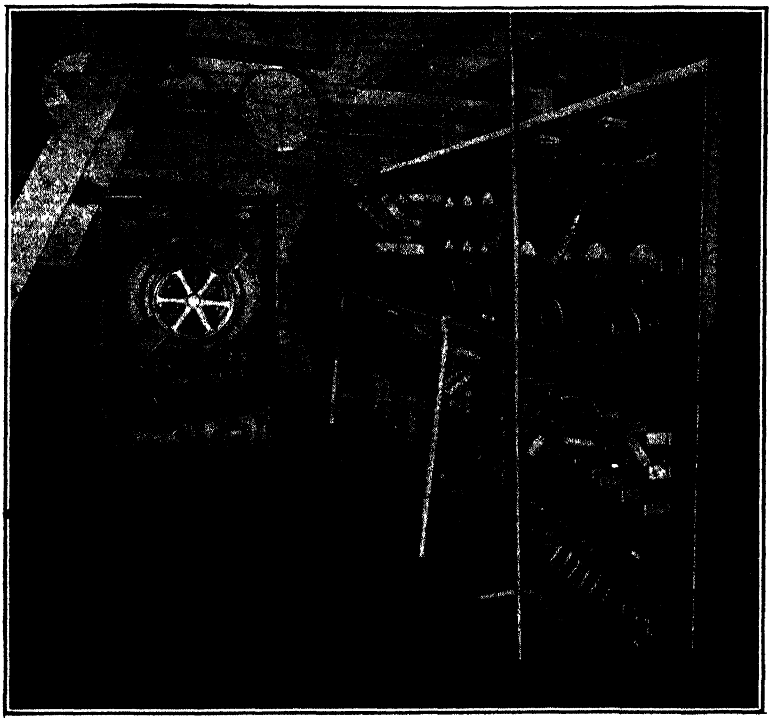


Fig. 309.—Engine Room Controlling Rheostat, Control Panel and Switchboard on Ferryboat "Golden Gate"

section of the starting resistor is placed in series with the motor armature, and on succeeding points other sections are cut in, in parallel with the first section, until on the last point a contactor closes short circuiting all the resistor sections for full speed operation. In stopping, the operation is simply reversed as the controller handle is moved towards the "off" position, and reversing is accomplished similarly by moving the controller handle from "off" to full speed in the opposite

direction, the current in the motor armature being automatically reversed when the controller handle passes through the "off" position.

The operating principle of Ward-Leonard control consists in controlling the speed of the motors by varying the voltage delivered to them by manipulation of the generator fields. Further refinement in control can be obtained by varying the

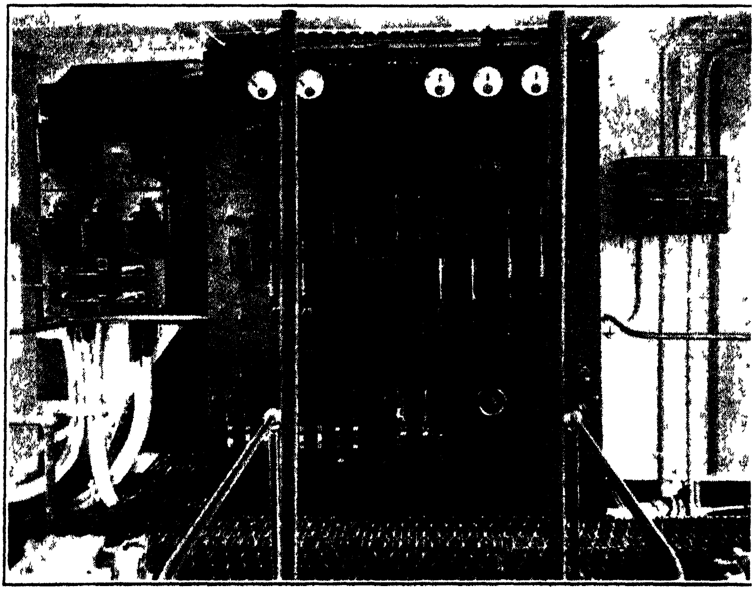


Fig. 310.—Front View of Main Control Panel, Tug Boat, "Van Dyke 3"

strength of the motor fields, and thus increasing or decreasing the torque per ampere for any particular speed.

The equipment furnished for a typical example of voltage control consists of an engine room control panel on which are mounted necessary switches, instruments, motor field rheostats, contactors and relays; a pilot house panel carrying instruments for indicating propeller speed, line amperes, and motor field amperes; a controlling rheostat for varying and reversing the shunt field current of the main generators, and which may be operated from either the pilot house or the

engine room, and an exciter control panel, mounted in the engine room. In addition, there are the necessary switches for disconnecting the motors and generators, transfer switches and protective devices.

The operation is as follows: After the main Diesel driven generators are brought up to full speed, with the controlling rheostat in the "off" position, the rheostat is moved to the

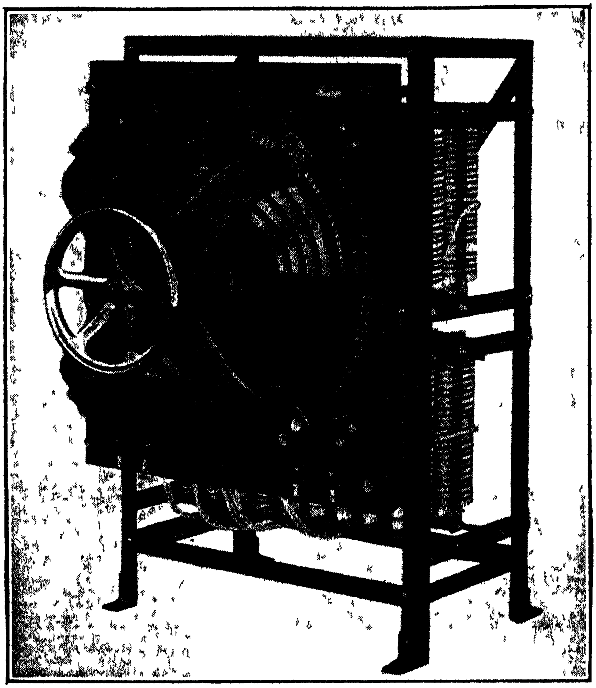


Fig. 311.—Rheostat Used with Ward-Leonard Control

first running position. This energizes the field of the generators, which supply current to start the motors. The field current of the generators is gradually increased as the rheostat handle is moved further along from the "off" position, until the motors are receiving full voltage, and are running at full speed. In stopping, the rheostat handle is merely moved in the opposite direction, back to the "off" position, and reversal

is obtained by moving it through the "off" position, and continuing in that direction. The rheostat, Fig. 311, is so constructed that continuous running is possible on any point; therefore, a large number of speed gradations are made possible.

Typical Installations. Up to the present time, the General Electric Company has furnished the electrical equipment for 21 Diesel-electric driven vessels.

The vessels thus equipped include representatives of several types which, aside from demonstrating the versatility of Diesel-electric drive, also serve as examples proving its ability to meet the special requirements of different sorts of marine service. In order to bring out the latter point without going into too much detail, a short description of one example of each general type of application with its requirements, and its salient features, is given in the following pages.

The circumstances which generally attend the operation of fishing trawlers tend to make the matter of operating costs the most vital factor of their existence. For this reason, if for no other, Diesel-electric drive, possessing as it does features that have a direct and beneficial effect on expenses, is particularly suitable for such service. The advantages of the drive which apply in this instance may be itemized as: First, reduced fuel expense; second, improved maneuvering ability, especially when navigating crowded harbors, and when docking; third, increased cargo space owing to the compact nature of the power plant; and fourth, the fact that the necessary auxiliaries, net hauls, hoists, pumps, etc., can be operated from the same source of power, and simultaneously with the main propulsion motors. The actual effect of these factors on the overall operating economy of a ship of this class has been thoroughly demonstrated in the case of the trawler "Mariner" which was equipped with Diesel-electric drive in 1920.

The "Mariner" is a wooden beam trawler, rated at 500 tons, with an overall length of 124 feet, beam 24 feet 3 inches, and mean draft 11 feet 8 inches. Her electric propulsion equipment consisted of two 165 K.W., 125 volt direct current, self-excited generators each directly connected to an eight cylinder, four cycle, 350 R.P.M., 240 B.H.P. Nelseco Diesel

engine. The generators, normally connected in series supplied current to one 400 H.P., 200 R.P.M. direct current compound wound motor direct connected to the propeller shaft. In addition to the main propulsion equipment there were several electric auxiliaries which, under cruising conditions were supplied with power from the main generators, and when the ship was in port, from an independent 15 K.W., 125 volt auxiliary generator driven by a small Diesel engine. The auxiliaries consisted of an emergency air compressor, motor driven bilge and water supply pumps, an engine room ventilating fan, a 65 H.P. double drum hoist for handling the net and a 5 H.P. whip hoist for unloading.

The control was rheostatic with stations in both the pilot house and engine room. Provision was made for manual operation in case of emergency.

The features of Diesel-electric drive that have most influence on the question of its application to cargo carrying vessels are practically the same as those mentioned in the case of trawlers, with perhaps greater emphasis placed upon the fact that increased cargo space is possible with Diesel-electric drive owing to the omission of shaft tunnels, and the small size of the prime movers. A further feature of advantage to cargo vessels is the uniform speed that is possible, whereby schedules are maintained with greater accuracy, and consequently more cargoes can be carried in a given time.

Up to the present, the General Electric Company has equipped cargo vessels of two different classes, the "Fordonian," a Great Lakes vessel, and the "Twin Ports" and "Twin Cities," designed for service, both on the Great Lakes, in barge canals, and for coastwise service along the Atlantic seaboard.

The "Fordonian," Fig. 312, is particularly interesting as she forms a means of direct comparison between direct Diesel and Diesel-electric drive, being converted from the former to the latter in 1922. It developed on her trial trip that both her speed and her maneuvering ability were greatly increased with the new form of drive.

She is 250 feet overall, with a beam of 42 feet, mean draft of 19 feet, and gross tonnage of 2,368 tons. Her propulsion

equipment consists of two 350 K.W., 250 volt compound wound, direct current generators, each driven by a two cycle, four cylinder, Ansaldo-San Giorgio 500 B.H.P. Diesel engine. The generators are normally connected in series, supplying 500 volts to a double armature, shunt wound, 850 H.P., 120 R.P.M., direct current motor, directly connected to the propeller shaft. The control system is combined rheostatic and Ward-Leonard, control being from the engine room. All engine room auxil-

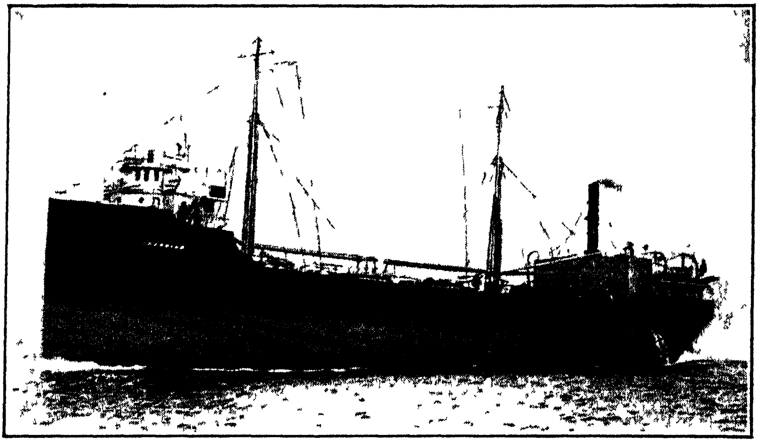


Fig. 312.—Motorship "Fordonian" Fitted with 850 H.P. Diesel-Electric Drive

iaries are electrically driven. A 15 K.W., 125 volt direct current auxiliary generator driven by its own Diesel engine is used to supply power for the auxiliaries when the main engines are shut down.

The "Twin Ports" and "Twin Cities" are sister ships, being exactly alike in dimensions and equipment. The dimensions are: length, 252 feet; breadth, 46 feet; mean draft, 9 feet 6 inches, and capacity, 3,000 tons. The electrical equipment consists of two Lombard Diesel engines driving two 250 K.W., 260 R.P.M., 230 volt compound wound direct current generators which supply power to two separate 250 H.P., 180 R.P.M., 230 volt shunt wound motors each directly connected to a propeller shaft. The control is rheostatic, and is arranged for

direct operation from the pilot house, with no intermediate control in the engine room, as is usually the case. The numerous auxiliaries may be operated either from the main propulsion motors, or by an engine driven 40 K.W., 230 volt auxiliary generator.

The requirements of tankers, as far as Diesel-electric drive is concerned, parallel those of cargo vessels. The General Electric Company has equipped four tankers, three of which, the "Standard Service," "Alaskan Standard" and "Hawaiian

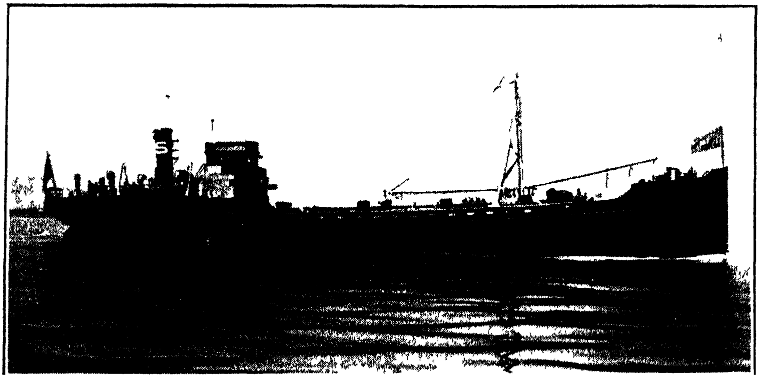


Fig. 313.—Tanker "Standard Service" Fitted with 600 H.P. Diesel-Electric Drive

Standard," are uniform as far as general dimensions and equipment are concerned, and the fourth, the "Brilliant," is slightly smaller.

The first three boats mentioned are all alike with the exception of the "Hawaiian Standard," which has her engine and generator room amidships and a separate motor room aft, instead of having her power plant and motors in the same room aft as is the case with the other two. This variation in the location of the component parts of the propulsion equipment shows how easily it can be arranged to suit the individual requirements of any case without sacrificing either space or operating efficiency.

These ships are practically uniform in dimensions, their length being 210 feet, breadth 40 feet, mean draft 15 feet, and

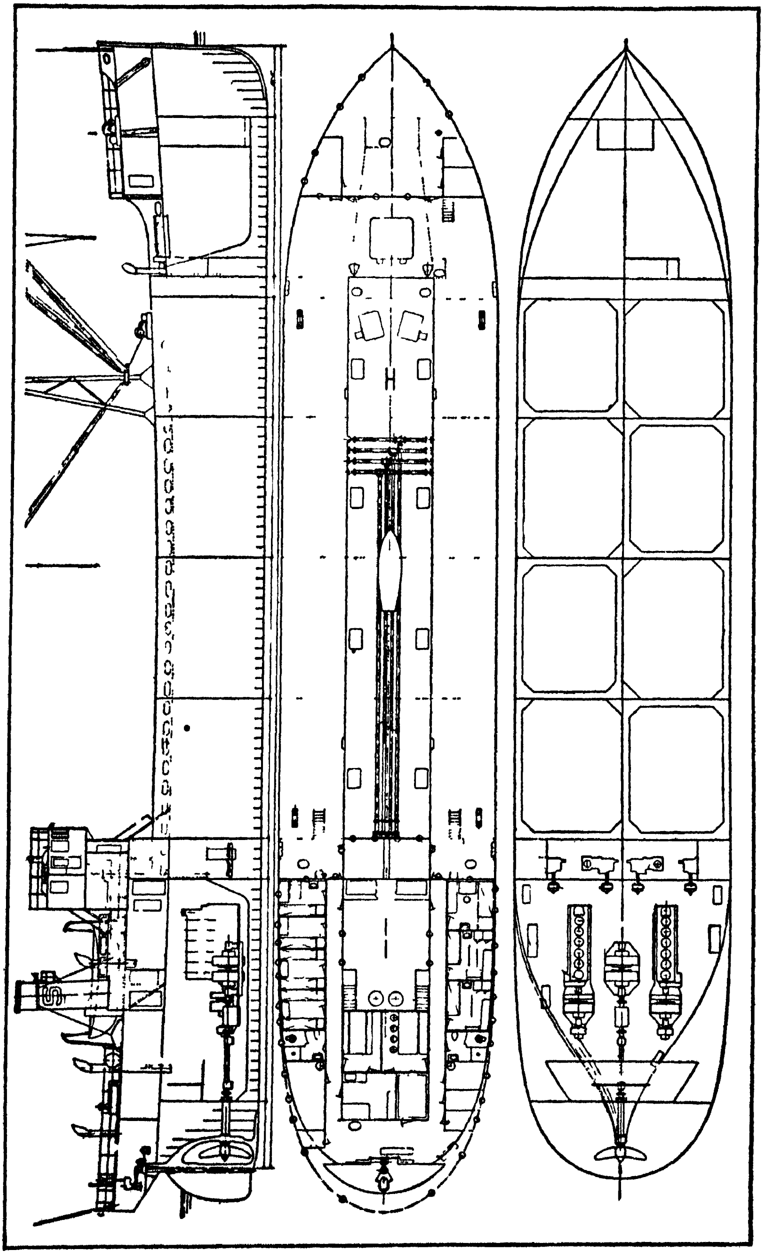


Fig. 314.—Plan and Elevation of Tanker "Standard Service"

displacement 2,725 tons. The propulsion equipment consists of two 4 cycle 400 B.H.P. Workspoor Pacific Diesel engines, each direct connected to a 245 K.W., 250 volt differentially compound wound generator supplying power to a 500 volt, 600 H.P. double armature shunt wound direct current motor, directly connected to the propeller shaft. There are also two 30 K.W., 115 volt exciters, direct connected to the generators, and an auxiliary generator used for lighting purposes.

The control is Ward-Leonard, so arranged that speed control and reversal is obtainable from either the pilot house or the engine room. An arrangement is also possible whereby the main propulsion generators can supply power for the cargo pump motors. The M.S. "Standard Service" is shown in Figs. 313 and 314.

The application of Diesel-electric drive to two different types of ferryboats (double ended and side wheel) has drawn attention to certain of its possible advantages that are especially valuable in such service. With both types of boat Diesel-electric drive results in a considerable saving in fuel consumption directly traceable to the elimination of standby losses, and to the fact that the increased maneuvering ability available makes docking at the end of trips a much more simple matter than with other drives. This maneuvering ability coupled with the positive and accurate control obtainable is a very valuable characteristic for ferryboats, operating as they do in crowded harbors.

Aside from these features, Diesel-electric drive has been the means of solving the old and troublesome problem that has confronted the operators of double ended ferryboats, namely, the concentration of the available power on the propeller that is actually driving the ship. Not only is this a simple matter where the propeller motive power is independent of the prime mover, as is the case with Diesel-electric drive, but it has been found that by running the bow propeller at a slow rate of speed, it is possible to save an amount of power at least equal to the losses in the electrical system.

With side wheel ferryboats, by using individual motors for each paddle wheel, either can be controlled, as regards speed and direction of rotation, independently of the other, and

without regard to the speed or direction of rotation of the prime mover. Since these boats are usually steered and handled to some extent by varying the speed and direction of rotation of the paddle wheels with respect to each other, this feature adds greatly to their maneuvering ability. Naturally, the same advantages of economy and accurate maintenance of schedules apply equally in the case of this particular type of boat.

The double ended ferryboats equipped by the General Electric Co. are the "Golden Gate," Fig. 315, the "Golden

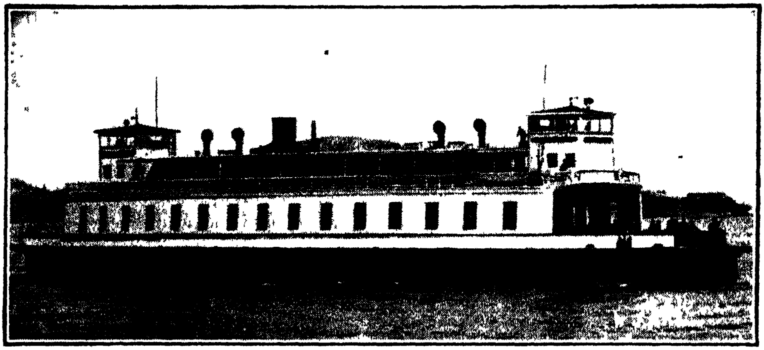


Fig. 315.—Ferry Boat "Golden Gate" Diesel-Electric Drive

West" and (in course of building) the "Golden State." The dimensions of the first two are, length 220 feet, breadth, 36 feet 6 inches, mean draft 11 feet 6 inches, and gross tonnage 598. The propulsion equipment which is the same for both, consists of two 500 B.H.P. Workspoor Pacific Diesel engines, each directly connected to a 360 K.W., 250 volt shunt wound, direct current generator, supplying power to two 750 H.P., 500 volt shunt wound direct current motors, one on each propeller shaft. Each of the generators is separately excited from a 35 K.W., 115 volt compound wound auxiliary generator mounted on the same shaft. Normally the generators are connected in series, supplying power to the motors, which have their armatures connected in parallel. Thus the proper speed relation between the two propellers is automatically secured.

The control is "Ward-Leonard" and may be operated from either of the two pilot houses or from the engine room, as desired.

The side wheel boat that has been equipped is a river ferry-boat, the "Froman M. Coots." Her dimensions are—length 172 feet, beam 45 feet, depth 7 feet and tonnage 490. Her propulsion equipment consists of two Fairbanks-Morse 2 cycle, 240 B.H.P. Diesel engines, each connected to a 175 K.W., 230 volt compound wound, direct current generator, which supply power to two 175 H.P., 230 volt, shunt wound direct current motors, each connected to one of the two paddle wheels through a system of double reduction gears. This system of double reduction is necessary to obtain the very slow paddle wheel speed of 14 R.P.M.

The control is rheostatic, with one station in the engine room and another in the pilot house, as shown in Fig. 316. Thus the pilot has complete control of the paddle wheels.

The application of Diesel-electric drive to towboats offers more possibilities in the way of improved operation and economy than almost any other class of ship. To anyone familiar with this class of vessels, and the work they do, the reasons underlying such a statement are readily apparent; nevertheless, the actual advantages arising from the use of Diesel-electric drive are worthy of some detailed consideration.

The principal advantages directly applying to towboat services may be summarized as follows:

The system permits the use of a propeller primarily designed for towing, as full power is constantly available when accelerating a tow, towing, or running light. It is also possible to adjust exactly the torque of the propeller to the existing circumstances, without necessarily altering the speed of the boat. In connection with economy, the low fuel consumption and absence of standby losses is an important feature, and, in addition, the same power plant that provides the energy for propelling the ship can be used for driving electric auxiliariės such as fire or salvage pumps, and cargo pumps located either on the towboat itself, on other vessels, or on shore.

One of the essential requisites of towboat service is the

ability to maneuver rapidly and accurately; this requirement is perfectly met in the case of Diesel-electric drive. Thanks to the accurate control available in the pilot house, the towboat and its tow can be controlled so as to meet any set of circumstances that may arise. For instance, in taking up the pull on the tow line, or in maneuvering a tow, the pilot of the

Fig. 316.—Pilot House of Diesel-Electric Side Wheel Ferry "Froman M. Coots"

towboat is able to watch the lines and his ship at the same time, and apply power gradually, and uniformly, as needed.

These statements are far from being theoretical, as they are thoroughly borne out by the results of actual comparative tests made on two similar vessels, one Diesel-electric and the other direct connected Diesel drive. Besides bringing out some unexpected virtues of Diesel-electric drive, they demonstrated its superiority over direct Diesel drive beyond any argument.

So far the General Electric Company has equipped two sets of tow boats, the "Van Dyke I, II, and III," and two

boats owned by the New York Central Railroad Company in course of building. The three "Van Dykes" are uniform as to dimensions and equipment. They are, overall length 97 feet; beam 21 feet; molded depth, 11 feet 6 inches. The propulsion equipment consists of two Price Rathbun three cylinder, 225 B.H.P. Ingersoll Rand Diesel engines directly connected to two 155 K.W., 125 volt shunt wound direct current generators, supplying power to a 370 H.P., 250 volt shunt wound, direct current motor directly connected to the propeller shaft. Excitation is furnished from either of two 26 K.W. shunt wound auxiliary generators mounted on the same shaft as the generators.

The control is Ward-Leonard, with operating station in the pilot house only.

The New York Central Boats are also uniform in electrical equipment. They are 108 feet long, beam 26 feet. The propulsion equipment consists in one boat of two Ingersoll Rand Diesel engines and the other McIntosh & Seymour engines of 400 B.H.P., directly connected to 270 K.W., 240 volt compound wound direct current generators which supply power to a 650 H.P., 480 volt double armature direct current motor directly connected to the propeller shaft. In addition to the two main generators, there are two 30 K.W. auxiliary generators for furnishing excitation, etc., mounted on shaft extension of the main generators.

The control is of the Ward-Leonard type, arranged for direct operation from the pilot house.

One of the newest applications of Diesel-electric drive is its use in furnishing power to self-propelled dredges. One of the principal advantages of the drive in this connection is that, since the motors operating the dredge pumps can be supplied with power from the main propulsion generators, when the dredge is not under way, the standby losses are reduced to a minimum. Of course, the other features of Diesel-electric drive, as to fuel economy, maneuvering, etc., apply.

The General Electric Company has furnished the electrical equipment for one twin screw dredge, the "Sandmaster." She is 261 feet long, 43 feet beam, and her draft when loaded is 18 feet. Her propulsion equipment consists of two Worthington

Pump and Machinery Corporation 600 H.P. Diesel engines directly connected to two 400 K.W., 230 volt, compound wound, direct current generators, each furnishing power to a 500 H.P., 230 volt, direct current motor, directly connected to its individual propeller shaft. One auxiliary generator furnishes excitation for the motors and generators, and another for engine room auxiliary apparatus, steering gear and lighting.

The main pumping apparatus, which is also driven from the propulsion generators, consists of two 400 H.P. pumps. There are also a large number of additional auxiliary pumps, windlasses, etc., which are driven from the same source.

The control is Ward-Leonard, with stations in both the pilot house and engine room, and providing independent control of each propelling motor.

CHAPTER XIV

Properties of Lubricating and Fuel Oils

Various Types of Oils—Distilling—Characteristics of Oils—Acidity—Ash Tests—Carbon Residue Tests—Oxidation—Surface Tension—Color—Emulsion—Effect of Cold—Effect of Salt—Density—Viscosity—Tests of Oils—Requirements of Lubricating Oils—Fuel Oils—Heating Value—Flash Point—Burning Point—Water Contents—Fuel Oil Testing.

Lubricating and fuel oils are obtained from crude mineral oils. These oils are distilled, and during the process of distilling, petroleum ether, gasoline, kerosene, fuel oil, etc., are distilled off. After these there remains lubricating oils, greases, paraffine, etc.

Oils are divided into classes chemically, that is, an oil is said to have a paraffine or an asphalt base. All crude oils are composed of a great number of hydro-carbons having hydrogen and carbon combined in different values. There are two classes of these hydro-carbons, one called the paraffine series and the other called the asphalt series. An oil is said to be of a certain base when the hydro-carbons of that series predominate.

The following are some of the characteristics and tests of lubricating and fuel oils:

Acidity. Acids in oils, in amounts greater than traces, cause rusting and pitting of the metal surfaces in which they come in contact, and will soon put shafts and bearings out of commission. A simple test for acids is to observe the effect on blue litmus paper. This is a qualitative test only, and for a satisfactory test a sample should be sent to a chemist for analysis.

Ash. All pure oils, when heated and ignited in the open air, burn quite completely, leaving almost no ash. Mineral soaps put into the oils leave ash, and the ash test therefore, becomes practically a test for the presence of soaps in the oil.

For internal combustion engine oils, the presence of ash is objectionable. Soaps usually raise the viscosity of oils and are used freely in greases, generally without objection. If the oil is to be used for lubricating very hot surfaces, however, the soap is liable to decompose and cause trouble by rusting and pitting metals.

Carbon Residue. When oils become very hot they give off volatile combustible gases and the hydro-carbons decompose partially. When this is carried to the end there remains a deposit called "Carbon Residue." Animal and vegetable oils and mineral oils loaded with resins and mineral soaps leave considerable carbon residue.

The carbon residue test has practically nothing to do with the amount of "carbonizing" by the oil in the cylinders and on the pistons, though it is often so taken. The conditions of a carbon residue test are entirely different from the conditions in an engine cylinder.

Oxidation. Exposed surfaces of oil gradually oxidize when brought in contact with air. The rate of oxidation increases with the temperature. The product of the oxidation is resinous or gummy, and spoils the oil as a lubricant if too much oxidation occurs. Mineral oils oxidize very little but the degree is much greater with animal and vegetable oils. An extreme case of oxidation is that of linseed oil which often oxidizes so rapidly as to cause spontaneous combustion. The action commonly called drying of paint is really the oxidation of the linseed oil. A good lubricating oil should be practically non-oxidizing.

Surface Tension or Capillarity. This, and the adhesion of oil to metal surfaces, are the forces which cause oil to spread over the metal, and crawl into the thin spaces between bearings and shaft, or piston and cylinder, while the metal parts are at rest.

Color. Some oil companies lay great stress on the color and specific gravity of their oils. The color of an oil has no relation to its lubricating value but results from the details of the manufacturing process. Mineral oils are naturally dark, or dark red, until the carbon and carbonaceous matter is removed by filtration, or bleached by sulphuric acid. About the

only information of any value that can be obtained from noting the color is whether or not it is a light oil compounded with a cylinder oil. If such is the case it shows the familiar greenish tinge of the cylinder oil.

Emulsion. A pure distilled mineral oil separates completely from water, which remains clear, within twenty-four hours. Acids, soaps, resins, etc., make emulsification easier, prevents the complete separation of oil and water, and leaves the water cloudy. There will often be, after the separation, a layer of foam between the oil and the water. The fact that an oil will or will not readily form an emulsion with water is often of importance in judging whether an oil will feed or not, as an oil pump suction will not take on the foam. The Texas Company states that it is impossible to break up emulsions formed by oil and water if there is present any oxide of iron, or iron rust, in the lubricating oil tanks.

Cold. Cold has the effect of rapidly increasing an oil's viscosity. During cold weather oils sometimes become so stiff that they will not flow at a readily measurable rate under the action of gravity.

With small engines lubricated by the splash system, it will generally be found the best policy to use a lighter oil in winter than in summer. There is very little difference in the working temperature of the cylinder walls in winter and summer, but the temperature of the crank case will show very great differences. If the oil is very viscous, due to low temperature, it will not splash very easily and too little oil will be supplied to the cylinders.

Salt. Salt water in lubricating oil causes rusting and pitting of the metal surfaces with which it comes in contact. The presence of salt can be determined by placing a few drops of silver nitrate in a glass, or test tube, of oil. If salt is present a cloudy, milky appearance is observed. In marine practice lubricating oil should be tested for salt at least once each watch.

Density. Density of oils has nothing to do with the lubricating value of an oil. Lubrication does not depend in any way upon density as a specific property of the oil.

When oils are spoken of as "light," "medium," or "heavy,"

the property concerned is viscosity. "Light" means low viscosity, "heavy" means high viscosity. What density does tell concerning an oil, seems to be the source of the oil—what kind of crude oil was used. Thus, densities above .90 indicates, in this country, that Texas crude oil was used as a base; densities below .90 indicates Pennsylvania crude oil as a base.

Density has an important commercial application when it is used to identify oils. If a new shipment of supposedly the same oil, that has been used, has very closely the same density at the same temperature as the original, it is practically certain that the new oil is the same as the old.

Density is expressed by an arbitrary measurement in degrees Baumé—the degrees into which the stem of a Baumé hydrometer is graduated. Density is measured in the laboratory by means of special accurate balances. It can be measured closely by simply weighing a bucket full of oil, and the same bucket full of water. Deducting the weight of the empty bucket, and dividing the net weight of the oil by the net weight of the water will give the density, or specific gravity of the oil.

The degrees of a Baumé scale are of a constant length, while those in a specific gravity scale grow smaller as the density increases. There is, accordingly, no simple relation between the two. The United States Bureau of Standards has determined that the readings on the Baumé scale may be converted to specific gravities by the following formula, in each case of which B is the readings on the Baumé scale:

(1) For liquids heavier than water—

$$\text{Specific Gravity} = 145 \div (145 - \text{B})$$

(2) For liquids lighter than water—

$$\text{Specific Gravity} = 140 \div (130 + \text{B})$$

To obtain the degrees Baumé from the specific gravity the following formula is used—

$$\text{Degrees Baumé} = \frac{140}{\text{Sp. Gr.}} - 130$$

The hydrometer is an instrument used for determining the densities of liquids, such as water and oil. It consists of a stem near one end of which are fixed two bulbs. The upper

bulb is the larger and is filled with air to provide buoyancy. The lower bulb is weighted to maintain the instrument in an upright position. When placed in a liquid the hydrometer sinks until the weight of the liquid displaced is equal to the weight of the hydrometer. Therefore, the denser the liquid the higher the stem is above the surface of the liquid, and vice versa. In order to utilize this feature for obtaining

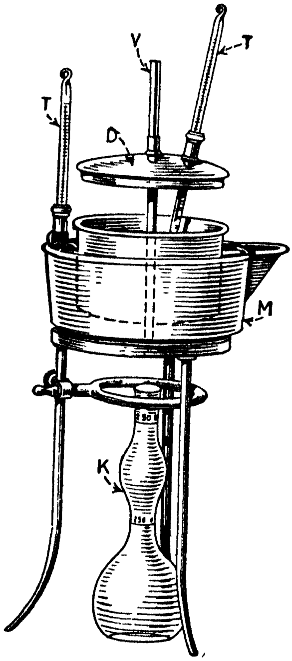


Fig. 317.—Engler's Viscosimeter

quantitative results, the stem of the hydrometer is graduated in accordance with some standard such as the Baumé.

Viscosity. Viscosity is a term applied to liquids to denote their consistency. A thick sticky liquid has a high viscosity, while a thin easily flowing liquid has a small viscosity. Viscosity is measured by means of the Engler's scale.

Engler's Viscosimeter, shown in Fig. 317, consists of a water jacketed chamber holding 240 cubic centimeters of oil. A flask, K, holds 200 cubic centimeters, which is the amount

being tested. The oil to be tested is placed in the receptacle D, and is brought to the required temperature by heated water in the water jacket M. This temperature is usually 60 degrees F.

If water requires 52 seconds for the passage of 200 cubic centimeters through to the flask K, and the same amount of

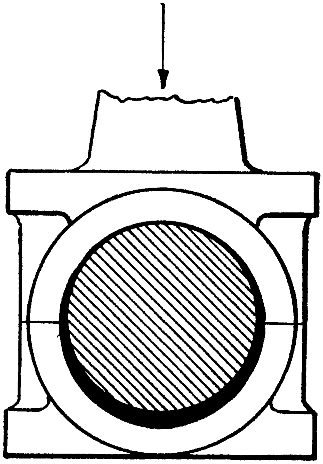


Fig. 318.—Shaft at Rest

oil under examination requires 130 seconds, then the ratio is determined by $\frac{130}{52} = 2.50$, the oil thus having a viscosity of 2.5 that of water.

Viscosity is the most important property of oils as regards lubricating qualities. It is practically the controlling factor, so far as the oil is concerned, in the formation and maintenance of the oil film between metal parts when the parts are in motion. Viscosity is the internal friction of fluids, the force resisting the movement of one layer of oil past the next. A liquid which, like steam engine cylinder oil or tar, is "thick" and does not flow readily, has a high viscosity.

When a crank brass is at rest on the crank journal, the shaft and bearing are in metallic contact on the side towards the load. The shaft, being smaller in diameter than the inside of the bearing, forms a crescent shaped space between the

shaft and bearing as shown in Fig. 318. This space will fill with oil, while the journal is at rest, by the action of surface tension, or capillarity.

When the shaft starts to turn, oil sticks to the journal and tends to carry around and form a film of oil between the bearing and journal on the side towards the load. The bearing, because of the load, tries to wipe the oil from the shaft, or to prevent the oil from going through. The oil layer in contact with the bearing metal stays stationary in any case. The

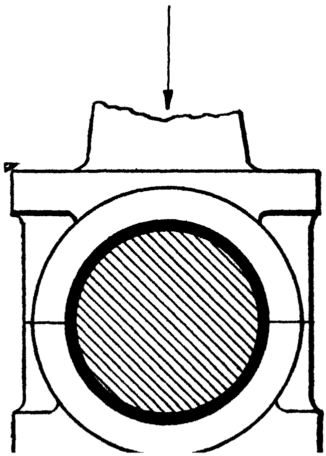


Fig. 319.—Shaft in Motion

resulting action is a motion of oil layer on oil layer, a motion which brings in the forces due to the viscosity of the oil.

When this viscous force, coupled with the wedgelike action of the oil under the pressure of the revolving shaft, becomes large enough, the oil film rather suddenly carries through, and the bearing and journal take the relative positions shown in Fig. 319. The oil film, of course, will always be thinner on the loaded side than on the unloaded side, but with good lubrication the shaft comes appreciably near to running concentric with the bearing.

Requirements of Lubricating Oils. A good Diesel engine oil, according to the Texas Oil Co., should fulfil the following requirements:

1. The oil must not congeal at any temperature to which it will be exposed in service.

2. It must readily spread over the cylinder walls, and not remain in streaks or patches.

3. The oil must be viscous enough to form a fair piston seal, but not so viscous as to cause undue fluid friction in the bearings.

4. It must not form bad emulsions as it is frequently mixed inadvertently with a little water in service, and in marine practice, sometimes even with salt water.

5. When exposed to the intense heat of combustion, which will decompose any oil, it must leave a minimum of carbon when distintegrated, and the carbon must not be gummy.

6. It must not be so volatile that an undue quantity will be necessary to maintain a good film on the cylinder walls.

7. It must not react and thicken with sulphur from the fuel.

For practical reasons it is desirable to use the same oil for all parts of the engine, that is, for both cylinder and bearing lubrication, and also for piston cooling if the engine is so equipped. Little need be said about the kind of oil to be used, in spite of all the elaborate specifications that have been so frequently published.

The operator, of course, should not use any oil that is not known to be well refined and of the first quality. The degree of refining is very difficult to judge by those who operate the engines, and they must accordingly rely on the reputation of the refiner.

Fuel Oils. Fuel oil should be free from all mechanical impurities of an organic or inorganic nature. When present they should be carefully removed by filtering. Clean oils are sometimes contaminated with impurities when loaded into tank cars in which dirt has been allowed to accumulate or from drawing oil from near the bottom of large storage tanks in which the impurities may have settled at the bottom during long periods.

Desirable fuels for Diesel engines should have the following properties:

1. It should burn completely without leaving any carbon

residue in the cylinder, and not clog up the exhaust valves and passages with soot, coke and ash.

2. It should be free from all mechanically held impurities which clog up strainers and score the plungers and pump barrels of the fuel measuring pumps.

3. It should be sufficiently fluid at ordinary temperatures to readily flow to the fuel pumps.

4. It should be free from water, as water lowers its heating value and may prevent its ignition.

5. It should be free from highly volatile oils, which evaporate and cause inflammable mixtures with air.

6. It should have a high heating value.

Heating Value. The heating value of an oil is important as it shows the energy available for power generation. This heating value is measured in British Thermal Units and should at least equal 18,000 B.T.U.'s per pound.

The fuel consumption per brake horsepower hour, for any given size engine and load condition, varies directly in inverse proportion to the heating value per pound of the fuel used; for example, if the fuel consumption per brake horsepower hour was .45 pound when using fuel having a heating value of 18,500 B.T.U.'s, then the fuel consumption for the same engine and same load conditions would be .462 pound per brake horsepower hour when using an oil of 18,000 B.T.U.'s per pound.

Flash Point. The flash point is the temperature at which an oil gives off inflammable vapors, which, mixing with air, forms an explosive mixture and burns on being ignited and then dies out. The flash point should not be lower than 60 degrees C.

Burning Point. The burning point of a fuel oil is the temperature at which it ignites and continues to burn in an open cup.

Several forms of laboratory instruments are in use for determining the flash and burning points of oils. These points, however, may be quite accurately determined by gradually heating a vessel containing the oil, stirring the same, and keeping a thermometer in the oil to determine its temperature. By passing a burning match over the oil, a flash will

occur when the flash point is reached. In the same manner the burning point is determined by continued heating, so that the oil upon igniting, continues to burn.

The flash and burning point is of value only as an indicator of the fire hazard. The nearer the flash point to the burning point, if the flash point is low, the greater is the fire hazard. A low flash point and a high burning point indicates the presence of highly volatile oils mixed with heavy oils. The burning point is usually 10 degrees to 50 degrees C. higher than the flash point. If the flash point is sufficiently high to preclude the fire hazard, determination of the burning point is superfluous.

Water Content. Fuel oils containing small proportions of water in the form of emulsions will not necessarily cause failure in ignition, although it reduces the heating value of the fuel. The water of such emulsions often contain salts in solution, especially in marine practice, and these salts are crystallized out with the evaporation of the water, causing incrustations and wear of engine parts.

Mechanical entrained water, when it displaces the fuel in the fuel pumps will cause failure of ignition and if in sufficient quantities will cause the engine to stop.

CHAPTER XV

Marine Rules for Vessels Propelled by Diesel-Oil Engines

(LLOYD'S)

Construction—Air Receivers and Pipes—Metal Tests—Pumping Arrangements—Spare Gear—Periodical Surveys—Rules—Bedplates—Crankshafts—Intermediate Shafts—Thrust Shafts—Propeller Shafts—Trials—Engine Auxiliaries—Compressors—Cooling Pumps—Fuel Oil Transfer Systems—Fuel Oil Injection Systems—Starting Arrangements—Scavenging Systems—Injection Air Arrangements—Lubricating Oil Systems—Cooling Water Systems—Exhaust Piping.

Section 1—In vessels propelled by Diesel-Oil engines, the Rules as regards machinery will be the same as those relating to steam engines, as far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

Construction

Section 2—In vessels built under Special Survey and fitted with Diesel Engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel Engines being built under Special Survey, the distinguishing mark ✠ will be noted in Red, thus: ✠LMC or ✠NE.

3. In order to facilitate inspection, the plans of the machinery are to be examined by the surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 pounds per square inch, the diameters of the crankshaft of the main engines are not to be less than those given by the following formula:

$$\left.\begin{matrix}\text{Diameter of}\\ \text{crankshaft}\end{matrix}\right\} = \sqrt[3]{D^2 \times (AS+BL)}$$

where D=diameter of cylinder,

S=length of stroke,

L=span of bearings adjacent to crank, measured from inner edge to inner edge.

The value of (AS+BL) are as given in the following table:

Table I

Four-cycle single acting engine cylinders	Two-cycle single acting engine cylinders	Values of the coefficients
4 or 6	2 or 3	0.089S+0 056L
8	4	0.099S+0.054L
10 or 12	5 or 6	0.111S+0.052L
16	8	0.131S+0.050L

For auxiliary engines of the Diesel type the diameters of the crankshaft may be 5 per cent less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should be not less than 1.33 times and the thickness not less than 0.56 times the diameter of the shaft as found above, or, if these proportions are departed from the webs must be of equivalent strength.

9. The diameter of the intermediate shaft must not be less than that given by the formula:

$$\left.\begin{matrix}\text{Diameter of inter-}\\ \text{mediate shaft}\end{matrix}\right\} = C\sqrt[3]{D^2 \times S}$$

where D=the diameter of cylinder,

S=the stroke of piston,

C is a coefficient found from the following table by interpolation from the values found for A.

Where the stroke is not less than 1.2 times, nor more than 1.6 times the diameter of the cylinder, (0.735D+0.273S) may be taken instead of $\sqrt[3]{\times S}$.

Table II

Two-cycle single acting engines. cylinders	Values of the Coefficients C where A=0.0025	A=0.0050	A=0.0100
2	0.305	0.317	0.336
3	0.346	0.363	0.385
4	0.364	0.380	0.396
5	0.380	0.391	0.404
6	0.398	0.403	0.412

Four-cycle single acting engines. cylinders	Values of the Coefficient C where A=0.0025	A=0 0050	A=0.0100
4	0 300	0.312	0.327
6	0.338	0.355	0.370
8	0.357	0.366	0.376
10	0 376	0.382	0.389
12	0.394	0.398	0.404

In using the above table the appropriate value of A is found from

$$A \times W \times d^2 \times R^2 = D^2 \times S$$

where D=diameter of cylinder in inches,
S=stroke of piston in inches,
d=diameter of flywheel in feet,
R=revolutions of engines per minute,
W=total weight of flywheel in tons.

10. The diameter of the flywheel shaft must be at least equal to that of the crankshaft.

11. Where ordinary deep collars are used the diameter of the thrust shaft measured under the collars must be at least $^{21}/_{20}$ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

12. The diameter of the screw shaft must be not less than the diameter of the intermediate shaft (found as above) multiplied by $\left(0.63 + \frac{0.03P}{T}\right)$ but in no case must it be less than 1.077,

where P=the diameter of the propeller in inches,
T=the diameter of intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern

tube, as provided for in Section 11, paragraph 3, of the Rules for Engines and Boilers for Steam Vessels. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be 21/20ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

13. If the designed maximum pressure in the cylinders exceeds 500 pounds per square inch, the diameters of the shafting throughout must be increased in the proportion of

$$\sqrt[3]{\frac{\text{Maxim. press. in lbs. per sq. in.}}{500}}$$

14. Where the cylinder liners are made of hard close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $^1/_{15}$th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

15. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 pounds per square inch, and must be perfectly tight at that pressure.

16. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

17. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent above the designed maximum pressure in the cylinders and discharging where no damage can occur.

18. The air compressors and their coolers are to be so made as to be easy of access for overhaul and adjustment.

19. Where the fuel is injected into the cylinders by air pressure, the following conditions are to be observed:

In single screw vesesls, an auxiliary air compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the maneuvering gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin screw vessels in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

20. A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

23. In all vessels fitted with engines in which the lubricating oil is circulated under pressure a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler.

Air Receivers and Pipes

Section 3—(1) Compressed air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

(2) Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. Quality of Metal—if made of welded, or riveted steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must

have a tensile strength not exceeding 30 tons per square inch (Section 33, Par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent less than that required with boiler plates.

5. *Tensile and bend tests* are to be made from the material of *each* receiver. When they are welded or riveted, the tests may be made, and the thickness verified before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the surveyor before the ends are closed in, and at this time the surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formula: Maximum working pressure in pound per square inch

$$=\frac{C\times S\times(T-2)}{D}$$

for thicknesses of $5/8$ inch and above,

$$=\frac{C\times S\times(T-1)}{D}$$

for thicknesses below $5/8$ inch,

where S=minimum tensile strength of the steel material used, in

T=thickness of the material, in sixteenths of an in.

D=internal diameter of cylinder, in inches,

C=coefficient as per following table:

Coefficient—

77 for seamless receivers of thickness of $5/8$-in. and above.

69 for seamless receivers of thickness below $5/8$-in.

54 for welded receivers of thickness of $\frac{5}{8}$-in. and above.

48 for welded receivers of thickness below $\frac{5}{8}$-in.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T=\frac{D}{17}\times\sqrt[3]{P}$$

where T=thickness, in sixteenths of an inch,
D=internal diameter, in inches,
P=working pressure, in pounds per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates for pressures up to 300 pounds per square inch is to be tested by hydraulic pressure $1\frac{1}{2}$ times the working pressure, plus 50 pounds per square inch. Where higher working pressures are used, the test pressure need not be more than 200 pounds per square inch above the working pressure.

12. All receivers above 6-inch internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air compressor is fitted with a safety valve so arranged and adjusted that no greater pressure than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

15. Oil or air pipes subjected to high pressure are to comply with the Rules for steam pipes, Section 13, Clauses 7 and 16 (Rules for Engines and Boilers of Steam Vessels).

Pipes which are subjected to a working pressure up to 1000 pounds per square inch must be tested hydraulically to at least twice the working pressure to which they will be subjected, and those subjected to a higher working pressure than 1000 pounds per square inch to an hydraulic test of at least 1000 pounds per square inch above their working pressure.

Pumping Arrangements

Section 4—The pumping arrangements are to be the same as would be required for steam vessels of similar size and power, with the exception that no bilge suction need be fitted to the main engine circulating pump. In the cases of vessels fitted with water ballast, the water ballast pump must have, in addition, one direct suction from the engine room bilges.

General

Section 5—(1) All oil fuel pipes, tanks and their fittings must comply with the requirements of Section 49 (Rules for Steel Ships).

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for electric fittings.

Spare Gear

Section 6—The articles mentioned in the following list will be required to be carried, viz.:

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, etc., fitted to it.

In addition, one complete set of valves, valve seats, springs, etc., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Diesel engines.

1 complete set of main skew wheels for one main engine.

2 connecting rod, or piston rod top-end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 connecting rod bottom-end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.

1 set of coupling bolts for the crankshaft.

1 set of coupling bolts for the intermediate shaft.

1 complete set of piston rings for each piston of the main and of the auxiliary compressors.

1 half set of valves for the main and for the auxiliary compressors.

1 fuel pump complete for the main engine, or a complete set of all the working parts.

1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.

1 set of valves for the daily fuel supply pump.

1 set of valves for the water circulating pumps.

1 set of valves for one bilge pump.

1 set of valves for the scavenge pump, where lift valves are used.

1 set of valves for the lubricating oil pump.

1 bucket and rod for the lubricating oil pump.

A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.

Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

Periodical Surveys

Section 7—(1) The engines are to be submitted to survey annually, and in addition are to be submitted to a Special Survey upon the occasion of the vesssels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey

will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 37 of the Rules for Engines and Boilers of Steam Vessels.

2. Special Surveys—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz.:—All the cylinders, pistons, valves and valve gears, connecting rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes sea connections and their fastenings, are to be examined. The air compressors are also to be examined. The air receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this Section.

3. Annual Surveys—The whole of the parts of the engines which the engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The survey must include, for each main engine, the examination of at least two pistons, two cylinder covers and their valves, two connecting rods and their brasses, both top and bottom ends. Two of the main bearings and crankshaft journals, and one of the tunnel bearings. If these are satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air receivers cannot be examined internally they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

The pumps and air compressors must be examined and tried under working conditions. If found to be satisfactory, they need not be dismantled.

The maneuvering of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. Record of Survey—If the various parts mentioned in paragraphs 2 or 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. Survey of Screw Shafts.—The screw shaft is to be examined annually and drawn at intervals as provided for in Section 37, Clause 3 (Rules for Engines and Boilers of Steam Vessels).

20. General—The following requirements apply to all oil engines for propelling and auxiliary purposes. All machinery parts subject to stresses are to be of sound material and the fits and clearances in accordance with the best marine practice. The passages for cooling water and lubricating oil must be carefully cleaned of sand and scale. The main bearings and the reciprocating parts should be readily accessible and lifting eyes or gear are to be fitted in way of main bearings and cylinder covers. The nuts of main bearing and connecting rod bolts are to be secured by split pins or other efficient means.

Hand or power turning gear shall be provided for all oil engines. The engines for propelling the vessel are to be fitted with a governor or other efficient means to prevent the speeding of the engines to more than 15 per cent above the designed number of revolutions; propelling engines over 300 B.H.P. should be direct reversible. Closed crank case engines should have suitable provisions to prevent the accumulation of gas in the crank case.

21. Bedplate—The bedplate or crank case is to be of rigid construction, oil tight, and to be provided with a sufficient number of bolts to secure the same to the ship's structure. The structural arrangement for supporting and securing main engines are to be submitted for approval.

22. Cylinders—Cylinders, liners, cylinder covers, pistons and other castings subject to high temperature or pressures are to be made of the best grade of cast iron or equally satis-

factory material. Castings must be free from defects affecting their strength.

Cylinders using a compression pressure of over 400 pounds per square inch are to be fitted with relief valves set to not more than 1 1/3 times the maximum working pressure, and the valve discharge should lead beyond a point of danger to life or vessel.

23. Crankshafts—The minimum diameter of the crankshaft is to be determined by the following formula:

$$d = a\sqrt[3]{\frac{D^2PL}{f}}$$

d=diameter of shaft in inches.
D=diameter of cylinder in inches.
P=initial working pressure in pounds per square inch.
I=fore and aft length of crank over webs plus 1 (inch).
a=factor from table below.
S=stroke of piston in inches.
t=thickness of crankweb in line with axis of shaft.
w=width of crankweb perpendicular to axis of shaft.
f=7500 for Grade 1 forgings.
f=8000 for Grade 2 forgings.
f=6500 for cast steel.

The value of P given by the builders must be verified by the Surveyor from indicator cards during the full power trial of the engine. A set of full power indicator cards of the main engines to accompany the Classification Report.

Subsequent adjustments for the purpose of obtaining higher initial pressures are not permitted without the special approval of the Committee.

Values of "a" for Air-Injection Diesel Engines

No. of Cyl.		S/L Ratios							
4-cyl.	2-cyl.	0.7	0.8	0 9	1.0	1.1	1.2	1.3	1.4
1–2–4	1–2	1.17	1.19	1.22	1.25	1.28	1.31	1 34	1.36
3–6	3	1.19	1.22	1.25	1.28	1.32	1 35	1 38	1.41
8	4	1.20	1.24	1.27	1.30	1.33	1.37	1.40	1.43
12	6	1.22	1.25	1.29	1.32	1.36	1.39	1.42	1.45
16	8	1.25	1.29	1.33	1.36	1.40	1.44	1.47	1.50

Values of "a" for Explosive-Combustion Engines

No. of Cyl.		S/L Ratios							
4-cyl.	2-cyl.	0 7	0 8	0 9	1 0	1 1	1 2	1 3	1 4
1-2-4	1-2	1 17	1 17	1 17	1 17	1 17	1 17	1 17	1 17
3-5-6	3	1 17	1 17	1 17	1 17	1 19	1 20	1 22	1 24
8	4	1 17	1 19	1 21	1 23	1 25	1 28	1 30	1 32
10-12	5-6	1 18	1 20	1 23	1 25	1 28	1 31	1 33	1 35

NOTE.—The above constants are to be used in cases where a bearing adjoins each side of each crank and where single impulses occur at equal intervals. In cases of departure from these conditions and for designs not properly coming under the two groups above, data must be submitted for the determination of shaft dimensions.

The dimensions of the crankwebs of solid shafts are to be such that wt^2 is not less than $0.4d^3$ and w^2t not less than d^3; for built-up crankshafts it is to be not less than 0.66d and w not less than 1.9 diameters of the hole in the webs. These proportioned dimensions are based on the use of the same grade of material for both shaft and webs and should be modified in accordance with the difference in the grade of the materials.

The webs are to be shrunk or forced onto the shaft and are to be fitted with dowels or keys of ample proportions to transmit at least 60 per cent of the torque. It is strongly recommended that the radius of the fillets in way of main bearings and the crank pins of solid shafts be not less than 0.05d. The shearing stress in the coupling bolts should not exceed the torsional stress in the shafting.

24. Intermediate Shafts—The diameter of the intermediate shafts is to be determined by the following formula:

$$d = b\sqrt[3]{\frac{D^2PS}{f}}$$

Values of "b"

Air-injection Diesel engines, number of cylinders			Explosive combustion engines, number of cylinders		
4-cycle	2-cycle	b	4-cycle	2-cycle	b
1–2–3–4–6	1–2–3	0 97	1–2–3–4	1–2	0 87
8	4	1 00	5–6	3	0 94
12	6	1 04	8	4	1 00
16	8	1 10	10–12	5–6	1 05
	12	1 20		8	1.10

Shafting for Auxiliary or indirect-drive engines may be 5 per cent less in diameter than required by the above formulae.

25. Thrustshafts—To be 5 per cent larger in diameter than the intermediate shafts.

26. Propeller Shafts—To be the diameter of the thrust shafts multiplied by the value of C in the table for Propeller Shafts in Sec. 34, Par. 3. The ratio in this case being that of the diameter of the propeller to the diameter of the *thrust* shaft.

27. Trial—Before final acceptance all engines must demonstrate by a running test their ability to perform satisfactorily the work for which they are intended.

28. Recommendation—In order to minimize operating troubles it is recommended that a system of periodical inspection, cleaning, replacement and adjustments of essential engine parts be followed by the ship's engineers.

Engine Auxiliaries

29. General—The following auxiliaries are intended to cover the minimum requirements for seagoing vessels of average power and should be increased for ships of large power but may be modified for River and Harbor Boats and for sailing vessels fitted with auxiliary engines. Propelling engines of a type and design not requiring certain auxiliaries noted below are to be provided with the necessary auxiliaries or connections in duplicate.

Pressure gauges, thermometers and relief valves are to be fitted where required.

30. Fuel Oil Transfer Pumps—One attached pump for each engine and one independent pump, or two independent pumps.

31. Compressors for Injection Air. Single Screw—One attached or one independent compressor of maximum capacity, and one independent compressor of 66 per cent capacity.

Twin Screw—(a) One attached or one independent compressor per engine of maximum capacity, and one independent compressor of 66 per cent capacity for one engine.

(b) One independent compressor of maximum capacity for both engines and one compressor of 66 per cent capacity for both engines.

(c) One attached compressor for each engine, capable and arranged to supply both engines for maximum requirements and one starting air compressor of adequate size.

NOTE.—The capacities noted above refer to the maximum requirements at any speed.

32. Emergency Air Compressors—One power operated compressor not requiring air for starting; capacity depending on size of propelling units.

33. Scavenging Pumps—One or more attached pumps for each engine or one independent pump. Crankpit scavenging will be approved for explosive combustion engines.

34. Main Lubricating Oil Pumps—One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements. Not required in cases of general multi-feed lubrication.

35. Water Cooling Pumps—One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements.

36. Note—Independent pumps denote pumps driven independently of the propelling engines. Attached pumps denote pumps driven by the propelling engines.

Engine Piping

37. Fuel Oil Transfer System—The piping arrangements for the carriage of fuel oil are to be in accordance with the requirements of Section 31.

The service tanks are to be located sufficiently high to permit gravity flow to the service pump suctions and shall be fitted with air vents leading to the atmosphere, drain cocks and a well-protected oil gauge with valves on both ends. Tanks are to be placed in drip pans provided with a drain to the oil drain well.

The pumps used for the transfer of oil are not to be used for bilge or ballast purposes; the suction pipe is to be fitted with an efficient strainer.

38. Fuel Oil Injection System—The suction to the fuel oil injection pumps is to be fitted with a Duplex Strainer and cut-out valves are to be located at the service tanks operated from the engine room floor. The piping in the discharge line should be of seamless drawn steel and the fittings of extra heavy steel; means should be provided to discontinue the fuel supply to each individual cylinder. The joints in the pipe lines should be metal to metal or have metal gaskets; ample provision is to be made for expansion. Individual injection pumps for each cylinder are recommended. Overflows and drip pans are to have drain pipes leading to the oil drain well.

39. Starting Arrangement—The efficiency and the capacity of the starting arrangement is to be demonstrated to the Surveyor in attendance.

On vessels fitted with Surface Ignition Engines all reasonable safeguards must be provided to prevent fires.

40. Scavenging System—The air supply is to be adequate for all speeds and the flow of the air should be regulated to insure uniform supply. The suction and discharge valves of scavenging pumps shall be readily accessible for examination and repair. Provision should be made to take care of excessive pressure by means of relief valves or breaking plates.

41. Injection Air System—The discharge lines of each stage of the air compressors are to be fitted with air coolers and relief valves of ample proportions. The temperature of the air discharge from each cooler should not exceed 150°.

A stop valve is to be fitted in the branch line to each cylinder so located that the engine may be operated with one or more cylinders cut off. The pipe lines may be made either of

seamless steel or seamless copper and should be fitted with drains.

The installation of oil separators is strongly recommended.

42. Lubricating Oil System—The lubrication of main bearings, crankpins, crosshead or wrist pins should be by means of pump pressure or by gravity. The pistons of the engines, compressors and scavenging pumps should be lubricated by individual pumps, adjustable to the particular requirements.

Extreme care must be taken to prevent the contamination of the oil supply by sea water, escaping oil gases or carbon. The oil drawn from the crank pit or sump must run through a duplex strainer, readily accessible for cleaning. The installation of oil filters or separators is recommended.

The tanks for the storage of lubricating oil must not form part of the ship's structure.

The flashpoint of the lubricating oil should in no case be below 350°F. In order to insure the continued safe operation of the engines, only the best quality of lubricants should be used.

43. Cooling-water System—The cooling-water pumps are to have at least one high and one low sea suction; the suction line is to be fitted with duplex strainers. For any emergency connection from the fire main the cooling-water should be led through the duplex strainer. Means should be provided to ascertain the temperature of the return from each cylinder and the proper circulation through all water jackets.

Drain cocks must be provided at the lowest point of all jackets and the discharge must be led to the bilge. A relief valve should be fitted in the main line to the jackets to prevent excessive pressure due to leaks.

44. Exhaust Piping—The exhaust pipes should be water jacketed or efficiently insulated. Where a number of cylinders are connected to one exhaust pipe, allowance should be made for expansion. Exhaust pipes of several engines should not be connected but run separately overboard, unless such connection can be made as will prevent the return of gases to an idle engine. Boiler uptakes and the engine exhaust lines should not be connected.

CHAPTER XVI

Two Hundred Diesel Engine Pointers*

The distinguishing feature of the Diesel engine is that the air charge is compressed to a high pressure such that the resulting temperature of the air is above the self-ignition point of the fuel oil. As a consequence the oil when introduced into the cylinder is ignited by the high temperature. The fuel is introduced at such a rate that the pressure does not rise above the compression pressure. Originally Dr. Diesel contemplated introducing the oil at a rate that allowed the heat created by the burning fuel to just equal the work done on the piston. If this had been possible, the temperature during combustion would have been constant. As it is, the temperature rises while the pressure is either constant or experiences a slight drop. For this reason the Diesel engine is known as a constant pressure engine; i.e., heat admission at constant pressure. The exhaust or heat discharge is theoretically at constant volume, although part of the heat is thrown out during the exhaust stroke at constant pressure.

The compression curve as shown by a Diesel indicator diagram is approximately an adiabatic curve. This means that no heat is lost to the cylinder walls and that pressure P_2 at any point of the stroke with the cylinder volume, V_2, can be determined by the equation.

$$P_1V_1^K = P_2V_2^K$$

when P_1 and V_1 are the pressure and volume at any other known point. In this equation K is equal to 1.35. Actually some heat is lost to the cylinder cooling jackets. The temperature at the end of compression is approximately 1,100 deg. Fahrenheit when the engine first starts; after warming up the temperature is around 1,400 deg. Fahr.

The usual compression pressure carried on modern Diesels ranges from 475 to 525 pounds per square inch. The compression is higher than is necessary to burn the fuel but, as the piston rings wear, the leakage lowers this value a considerable amount. Furthermore, the cooling effect of the air blast as it blows the fuel into the cylinder chills the air charge in the cylinder. This may be as much as 300 deg. Fahr.; and so a temperature somewhat above the actual ignition point of the oil must be carried.

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In the first Diesel engine patented, air was not used to inject the fuel charge. This idea was a later development. It appears that compressed air serves to atomize the fuel particles in a very perfect manner. Outside of the chilling effect mentioned no better means can be found to act as the injection agent. The air pressure carried depends upon the load on the engine, the character of the oil fuel and the design of the atomizer disks and cone.

In determining the air pressure to be carried the engineer should follow the engine builders' instructions. As the oil is varied, it will be found necessary to alter the pressure. With heavy fuel the pressure should be around 75 atmospheres at full load, 60 atmospheres at ¾ load, and 55 atmospheres at ½ load. If the fuel is light, around 28 degrees Baumé, the pressure should be dropped 5 to 10 atmospheres below these values. At all times the pressure should be carried as low as it is possible to carry it and avoid a smoky exhaust. With an old and worn engine the compression may be as low as 325 pounds per sq. in. It is then possible to still carry full load by raising the injection air pressure above normal. With some engines, pressures have been observed as high as 100 atmospheres.

In erecting a Diesel engine the foundation should be thoroughly reinforced with old rails, reinforced bars, etc. The best concrete mixture depends on the character of the sand and gravel. A ratio of 1 part cement, 2½ parts sand and 3 parts crushed rock or gravel is well balanced. The concrete should not be too wet. It should not be rammed to any great extent. A rod should be used to push the concrete into the corners, etc. After finishing the concrete, the foundation should stand at least 10 days to ripen. During this process the surface is best covered with wet sand.

The engine base must be set level if the engine is expected to operate at all satisfactorily. It is never advisable to be satisfied with an approximate level position. To line the base, iron wedges can be driven in between the base and the foundation.

The crankshaft-bearing housings on all modern engines are bored true to within 0.001 inch. The bearing shells are bored to size so that the crankshaft must be level if the base is level.

In the manufacture of the modern oil engine the bearings are made up of babbitt-lined semi-circular shells. These shells rest in housings formed in the engine frame. It is the practice to bore all the housings on a horizontal boring mill at one setting. They are then checked by a special fixture. The work is usually performed in a painstaking manner, and seldom does any housing show more than a few ten-thousandths

out of alignment. The housings are filed and scraped until they test true. The bearing shells are all made of equal thickness and, after being bedded in the housings, are checked for alignment by a test shaft. If any error appears, it is corrected by scraping the babbitt. After the engine is installed in a power plant, the bearings begin to wear. No matter how excellent the lubricating system may be, the wear starts as soon as the engine turns over. Since the babbitt linings in the several bearings are never of equal density the wear is by no means uniform. Varying pressures in the engine cylinders tend to increase the discrepancy in the rate of bearing attrition. If the flywheel weight is supported or partly supported by one of the end main bearings, this bearing will probably show a more rapid wear than the others. Oil-engine operators should recognize these facts and take proper steps to maintain the bearing alignment. Too many allow matters to drift until one or more bearings start to run hot. In most instances it is then too late to save the babbitt lining. Just as there should be a regular schedule in grinding valves and pulling pistons, the engineer should test the bearings at stated intervals. It is difficult to give a rule as to the length of time elapsing between inspections; the operator is not unduly active if he arranges to examine the bearings every six months. To remove the bottom half of the bearing, it is only necessary to lift off the top half and turn the flywheel. The revolving shaft will draw the lower bearing from the housing. The thickness of the shell should be measured at both ends and at the center; several measurements should be taken at various points. If these measurements are the same, it is evident that the wear on the particular bearing has been uniform. If the measurements check with readings taken on the other bearings, the main bearings are in alignment. If one bearing shows that it is low, the engineer must decide his course of action. It is possible to shim up under this low bearing until it is as high as the other. If the difference is marked this is the best procedure; sheet steel make the best liners. If the variation is less than five one-thousandths of an inch, probably the best scheme is to scrape the other bearings, removing an amount of babbitt to compensate for the variation between the highest and lowest bearings. In cases where all the bearings vary, scraping is the only remedy.

Most Diesel engines run best with a clearance of 0.004 to 0.006 inch between the shaft and top bearing shell. This should be checked at intervals by means of lead wires placed on the shaft before the bearing cap is pulled down. It is necessary to draw the cap snug. Upon removal of the thickness of the wire will indicate the clearance. If excessive, part of the shims should be removed. With pressure oiling systems the shims must fit the shaft very snugly at each end of the bearing.

In case a bearing burns out or is damaged a new one should be ordered from the factory. If the engine plant is equipped with a lathe

and other machinery, a perfectly good bearing can be made by the engineer. It is never permissible to attempt to run the bearing using the shaft as a mandril. The old bearing shell should be thoroughly cleaned and then tinned. The mandril can be made of steel or cast-iron at least ⅛ inch smaller in diameter than is the engine shaft. The bearing halves should be bolted together and placed on an asbestos pad and the mandril inserted. It is advisable to heat the mandril before pouring the babbitt. After the bearing is poured, it should be placed in a lathe and bored out to size. In reboring the bearing care must be exercised to get the bore square with the machined end of the bearing shell.

In rebabbitting any oil engine bearing old babbitt should never be used. Always use fresh metal and see that it is a high grade metal. It should never be overheated; if a pine stick chars upon being thrust into the molten metal, the babbitt is at about the right temperature.

Many oil engine crankshafts display a tendency to shift back and forth lengthwise as it rotates. If this is not stopped, the connecting-rod bearings will "bell" very quickly; often the cylinder walls are scored by the side thrust of the piston and rod. The placing of a steel or brass plate between the crankthrows and main journal will often stop the shifting, or babbitt rings can be run on the bearing shell ends. This latter is the better method.

In determining the clearance between the crank pin and big-end bearing the two bearing halves should be drawn up against the crank pin. The connecting-rod should be unshipped. Then if the bearing just grips the pin, it is presumed that the bearing and pin has no clearance. Measuring the distance or clearance between the two bearing halves, this distance plus the desired pin clearance of 0.004 to 0.006 inch gives the amount of shims required between the bearing halves.

In fitting a bearing shell to the shaft, the shaft should be lightly coated with Prussian Blue or lamp black. The bearing is placed upon the shaft and gently revolved several degrees. The high spots, upon removal of the bearing, will carry a streak of the lamp black. Scraping these spots, after repeated trials, will bring the entire bearing to a perfect contact. If a bearing has been bored true, there need be but a little scraping performed. After the lower shell is slipped into place, it should be removed to see if the shaft contacts along the entire bearing length.

In cutting oil groves, these should never extend to the outer edge of the bearing. If this precaution is not observed, the lubricating oil escapes entirely too fast and does not lubricate the shaft as it should.

Often the big-end bearing of the connecting-rod "bells." The causes of this, other than side-shifting of the shaft, are unknown. Probably in most cases the piston-pin bearing and crank pin bearing are not in line. If the rod does not move true with the center line of the piston, at least one of the pin bearings must wear unevenly, thus "belling."

The piston frequently slaps very violently. This can be and is caused by a worn piston, especially if the rod bearings are not square with each other. At other times mere excessive piston wear will produce a decided slap. New pistons or cylinder liners then become necessary.

If a piston is badly worn, the use of new piston rings will serve to hold the compression, but they are but makeshifts. If a piston is so badly worn as to "slap," it is safe to assume that a new piston is necessary. Modern pistons are usually made with a taper at the top starting below the bottom rings; the clearance close to the top may be ⅛ inch. At the belly of the piston the working clearance is usually 0.008 inch on vertical trunk pistons up to as much as 0.0015 inch per inch of piston diameter on horizontal crosshead type pistons. Often the skirt below the piston pin is smaller in diameter than the body of the piston.

Pistons distort due to expansion of the piston pin and score the cylinder liner. If the scores are not deep, the liner may be smoothed with a fine emery stone, finishing up with an extra fine emery paper. If the score is halfway down the cylinder, there is no danger of loss of power even if the scores are very deep.

Piston rings often gum very badly. This is due either to too much lubricating oil or to imperfect combustion. The latter may be produced by overload, low air injection or too low a compression pressure. In removing the piston rings care must be used and the rings not expanded more than necessary, or the rings may break.

If a piston ring sticks in the groove, the gum can be loosened by using a piece of spring brass. Copious supplies of kerosene will also serve to loosen the ring; lye water is equally serviceable. The groove and ring should be thoroughly cleaned before the ring is placed on the piston. If the ring is new, frequently the edges are rough and sharp. All roughness should be removed by a smooth file. In replacing piston rings, the ring should be fitted to the cylinder with about 0.01 inch gap clearance between the parted ends. The outside surface of the ring should be coated with lamp black, and then the ring is slipped into the cylinder. The contact between the cylinder and ring will remove the lamp black at all points of contact. By smooth filing the ring can be fitted to the cylinder. Proper fitting is necessary to prevent the escape of the gases across a point of noncontact.

Many engineers try to weld fractured piston heads. This is seldom, if ever, successful. The only feasible method of repair is that of "sewing" which is successful only in cases where the fracture is not of any great extent. Threaded brass rod makes the best kind of plugs to use in the sewing process.

If a piston head accumulates a lot of carbon, the compression is too low, or the injection air pressure is low, or the amount of lubricating oil is excessive.

In grinding the admission or suction valve and exhaust valve a ground surface ⅛ inch wide is ample. If the valve is ground excessively, the surfaces of the valve and cage will convex so that the sealing contact is poor. While some engineers mix up their own grinding paste, prepared compounds are better and can be secured at any automobile supply house.

In grinding very little pressure need be placed on the valve. The weight of the valve alone is sufficient to hold the rubbing surfaces together. The work valve should be given a back-and-forth motion covering about half a circle; then the valve should be lifted and moved 180 degrees and the grinding continued.

Many Diesels "pink" or detonate with a change in the fuel valve setting. This detonation is in every respect an explosion rather than a slow burning of the fuel as should occur. It has been found that the shorter the period of fuel valve opening the greater the tendency to detonate. Strangely, tests have proven that the heavier the engine detonates the greater is the load that can be carried. The fuel consumption with heavy detonations is lower than otherwise; this is to be expected since the efficiency of the Diesel depends, among other factors, on the length of the period of fuel introduction.

Examination of a series of tests on Diesel engines show that the difference between brake horsepower and indicated horsepower is fairly constant at all loads. This is shown as follows:

	b.hp.	b.hp.	Difference	Mech. Eff.
Full load	100	72	28	.72
Three-quarters load	82	54	28	.66
Half load	64	36	28	.56
One-quarter load	46	18	28	.39

The total amount of heat supplied by the fuel is used up in the following manner:

Heat generated in cylinder	100 per cent
Heat converted into work	30 per cent

Heat lost in engine friction	6 per cent
Heat lost in exhaust gases	28 per cent
Heat lost in cooling water	34 per cent
Heat lost by radiation	2 per cent

The Diesel excels the steam engine plant in thermal efficiency at approximately the ratio of 3½ to 1. The heat losses in a steam plant average as given below:

Total heat in the coal..............................	100 per cent
Heat lost in grate radiation	9 per cent
Heat lost in stack	22 per cent
Heat lost in engine-pipe radiation	2 per cent
Heat lost in engine exhaust	57 per cent
Heat turned into work...............................	10 per cent

For this reason the steam engine cannot compete with the Diesel unless the exhaust steam can be used in heating, etc.

If a main journal or crankpin becomes scored, it can be remedied by careful lapping with a canvas strip coated with emery paste. If it persists in running hot, white lead coated over the surface will often smooth it up.

With a pressure oiling system the engineer often thinks the reason a bearing warms up is because of insufficient oil grooves. Actually the groove does not assist in the lubrication; it merely furnishes a way of escape for the oil without doing work. If a bearing runs hot, as a rule the oil is feeding too rapidly to another bearing, resulting in a deficiency at the hot bearing.

Each make of engine differs as to the number of oil feeds to the piston. The majority use at least two, some as high as four. Two are ample. The rate of feed should be from 10 to 20 drops per minute. On enclosed crank case type engines there is considerable lubricating oil in a fog in the frame. This serves to lubricate the pistons, cutting down the amount fed through the oil pipes. The amount of lubrication required by a Diesel varies with the type and size. On large vertical engines, it has been possible to secure 15,000 horsepower hours per gallon. A fair average is 6,000 horsepower hours, or a 600 horsepower engine should run 10 hours on one gallon.

Lubricating oil used on the main and crankpin bearings will pick up minute particles of carbon if the engine is of the trunk piston design. This carbon will not settle out the oil, and a first class filter or filter press is necessary. The loss in oil should be made up at least once a week; the system should be kept filled. It does not pay to use cheap oil. All reputable oil companies have an oil suitable for Diesel

engines. The engine builder has tested all these oils and is in a position to advise as to the best oil.

In cleaning pistons a brass scraper should be used; emery cloth is never advisable save in cases of extreme scoring. Pistons should be pulled at least every nine months for cleaning. If the oil is a heavy residuum, more frequent cleaning will be necessary.

In adjusting the wear of the piston pin and crank pin bearings the length of the connecting-rod is reduced to a slight extent. To maintain the proper compression in the cylinder shims must be inserted between the rod and the big end to compensate for the reduction. While the engine will continue to function even though the clearance be materially increased and the compression pressure reduced, the greatest efficiency is secured at a certain compression dependent upon the engine design, character of fuel, etc.

In drawing up the big-end bearing bolts a heavy hammer should be used to sledge the nuts up tight. However, too much striking should not be indulged in, for the bolts may be unduly stretched by the sledging, resulting in a fractured bolt while in service. If the cotter key holes are considerably above the nuts, the intervening space should be filled with washers to prevent the nuts slacking loose.

Broken big-end bolts fracture in service at times. Some are caused by too great a stress in tightening the nuts. Most cases are due to stress occasioned by slackness or play in the crank pin bearing. If the play is excessive, there is a considerable stress from the inertia of the piston and rod at the dead centers. If the crank pin bearing pounds, it is necessary to take up the play at once. The usual wear should not exceed 0.01 inch in six months.

In visiting oil engine plants many Diesels ten to fifteen years old are often encountered. The majority of these old engines have never had a reasonable amount of care. Usually, the pistons are so badly worn that the compression is seldom above 300 lb. per square inch. On starting, the air blows down past the pistons so that it is difficult to turn the engine over fast enough to secure ignition. Many engineers, not realizing the danger, are in the habit of giving the cylinders a shot of gasolene. The gasolene is poured in through the admission valves. The engine, even though just barely turning over, will fire the gasolene in the firing cylinder. There is grave danger in this procedure. The gasolene will quite likely explode early in the compression stroke. If this occurs, something is going to give way. Even if the relief valves are fitted to the cylinder heads, they may fail to open. Frequently, admission valves cages are blown out, connecting-rods bent and crank-

shafts fractured. There is no disputing the fact that it is extremely difficult to start a badly worn engine. The best way to do it is to set the air starting valve to open after the piston has passed dead center. This will save a lot of air, since the piston will be moving outward when the air strikes it on the second turn. To prevent too much leakage about the pistons a pint of lubricating oil can be poured in on the top of each piston. The oil will effectually seal the clearance and will raise the compression pressure high enough to ignite the fuel when the fuel valve opens. There is no danger of the lubricating oil preigniting. The engine, even though badly worn, will usually fire. There are extreme cases of engine wear where even the method outlined above fails to secure results. It is possible to block open the admission valves of the firing cylinders and turn the engines over with the starting cylinder until a fairly high angular velocity is attained. Throwing the admission valve into place at this time will invariably secure regular firing. Of course, the proper relief is the reboring of the cylinder and the installation of new pistons. It so happens that occasionally the engineer cannot induce the management to do this. Above all else, no Diesel engine should be dosed with gasolene or kerosene.

With the closed-nozzle type of fuel valve the needle often scores the seat. Minor scores or cuts can be removed by grinding. If the valve seat is very rough, it may be repaired by rereaming. Frequently, the wear is so very rapid that constant reaming finally allows the tip of the needle valve to protrude and strike against the flame plate. It is impossible to fully determine what causes the rapid cutting. Some engineers, thinking the spring compression too high, have slacked back the adjusting screw. The valve then apparently cuts as fast as before. Evidently, if the spring resistance is not high, the needle valve chatters on the seat when closing. If the spring compression is high, the valve seats with a blow, but does not chatter. There is, then, a point in the adjustment of the spring where the cutting effect is at a minimum. This can be determined by trial. When the valve-cage seat has worn to such an extent that the needle valve tip protrudes, some engineers discard the cage and obtain a new one. At a small expense an engineer recently repaired several fuel-valve cages, making them fully as serviceable as new ones. When the valve functioned no longer, the cage was removed from the engine. The end was cut off, bored and threaded. A tip was turned out of steel and screwed into the cage. This repair enabled the operator to avoid the purchase of a new cage. The cost of repairing was about five dollars if the engineer's time be charged to the job.

To cause the several engine valves to function properly the clearances between the rocker arm rollers and cams must be correct. The manufacturer will supply the necessary data. Usually the valve gear will be noisy if the clearance is too great. In setting the fuel valve

the clearance should first be adjusted and the cam nose moved until the point of opening is correct.

In setting the point of fuel valve opening the indicator plug should be removed and the injection air, at about 300 lb. pressure, turned on. As soon as the air can be heard issuing from the valve when one's ear is placed at the indicator opening, the valve is starting to open.

In grinding the fuel valve a mere threadlike surface is necessary to secure sealing—a $\frac{1}{32}$ inch width is ample.

The engine should never be started until the air pressure is turned on to the fuel valves. If the air pressure is allowed to drop below the cylinder compression pressure, the flame may back up through the atomizer tip and ignite the oil in the fuel valve casting. A number of airline explosions have been due to this lack of care.

Special care should be used in seeing that the camshaft, or layshaft as it is sometimes called, bearing caps are tight. If any slackness exists in the bearings, the camshaft will give as the cam strikes the rocker arm roller. This is transmitted to the gears which will wear rapidly. On some Diesels the camshaft gears wear very fast. At times this is due to vibration in the engine crankshaft; in other cases it is caused by slack camshaft bearings; in other instances the wear is due to imperfectly cut gear teeth. Probably the latter is the most prevailing cause. It is well to remember that these gears need a copious supply of lubricating oil of a fair grade stock, such as steam cylinder stock.

While originally it was believed that there should be a fairly horizontal combustion line at constant pressure and all the Diesels gave such an indicator diagram, it is not the present practice in Diesel engine design to have a flat combustion line; a more peaked line is usual. A sharp combustion line gives a greater thermal efficiency; furthermore, even with a flat combustion line for some 10 per cent of the stroke, the actual combustion of the fuel occurs well along the stroke—as much as ⅝ stroke in many engines. The shape of the combustion line is dependent on the shape and lift of the fuel cam nose and on the injection air pressure.

The specific heat at constant volume in an oil engine is about 0.169. That is, if one pound of air is heated in a confined space (at constant volume) 0.169 B.t.u. are required to raise the temperature one degree Fahr. As the temperature rises, this specific heat changes, but for all practical purposes the value 0.169 may be assumed to be constant for all temperatures.

The specific heat at constant pressure, or the B.t.u. required to raise a pound of air, at constant pressure, one degree is 0.238. This is higher than the specific heat at constant volume, since there must be added to the constant volume specific heat the heat required to overcome the external work as the air increases its volume.

The air compressor of all save the small powered Diesels have three stages. The compression ratio per stage is usually 4 to 1. The temperature and pressure rise is then fairly low. If the intercoolers are effective there is no danger of an explosion of air and lubricating oil in any of the compressor cylinders. It is apparent, though, that if the intercoolers are clogged up or the water not flowing the air will reach a high temperature by the end of the final compression. With a 75 to 1 ratio and adiabatic compression with no loss of heat to the compressor walls the temperature T_2 at the end of the compression in the final stage will be

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

When n = 1.2 for air, T_1 = 80 degrees Fahr., or 540 degrees absolute; $\frac{P_2}{P_1} = 64$
then

$$T_2 = 540\ (64)^{1-\frac{1}{n}}$$

$$T_2 = 540\ (64)^{\frac{1.2-1}{1.2}} = 10800° \text{ absolute}$$

The temperature is much above the ignition point of many oils and is so high that the metal would melt long before the maximum pressure was reached.

The most prevalent cause of explosions in the air pressure lines is the sticking open of the fuel needle valve; this allows the flame in the engine cylinder to blow back through the fuel valve. The needle may be wedged open by lack of clearance between the cam nose and fuel rocker arm roller, or by tight packing or low injection air pressure; the first is probably the cause in most cases.

Many engineers are confronted with a decreased injection air pressure caused by leaky air compressor valves but are unable to determine which valve leaks. If a valve appears to be warmer than usual, it is proof that the valve leaks, providing no valve ahead of it is hot. In the latter case this last valve is the leaky one. As example, if the high pressure suction and discharge valves are hot as is also the intermediate discharge valve, the engineer can be certain that the intermediate valve leaks. If the high pressure discharge valve only is hot it is the leaky one. Another way to detect the leaky valve is by the

drop in the gage pressure on the engines having pressure gages on each stage.

If too great an amount of lubricating oil is fed to the compressor cylinders, the excess carbonizes on the valves, causing them to leak and heat. Considerable oil passes into the air line and bottles where it will form an emulsion with the moisture in the air. This emulsion gums up valves and air lines. The proper amount of lubrication depends on the design of compressor. As a guide two drops per minute per cylinder can be used in the absence of more definite manufacturer's instructions. If the cylinders have enough oil to dampen a cigarette paper when placed against the inside cylinder wall, the lubrication is ample. Many engineers do not lubricate the high pressure cylinder, depending upon the oil coming in with the air from the intermediate cylinder.

Due to the very small clearance between the high pressure piston and cylinder head the engineer must use care in taking up the crankpin and pistonpin play. In this adjustment the cylinder clearance should be measured and the rod adjusted accordingly.

Since the air drawn into the air compressor contains some moisture which is condensed by the cooling water in the intercoolers these intercoolers should be drained at frequent intervals. In some plants the drain cocks are opened hourly.

The connecting-rod bearings of the air compressor do fully as much work per square inch of projected area as do the main rod bearings. For this reason the former calls for as much care and attention as the latter.

In drawing up the engine cylinder heads the nuts should all be run up snugly. Then one nut should be tightened a part turn, followed by the tightening of the nut on the opposite stud. After all nuts are tightened fairly well, a sledge or heavy hammer should be used on each nut. All studs and nuts should be numbered so that the proper nut goes on its stud. If a nut turns down hard, it should be lapped with emery paste, so that it turns easily by hand. No nut should be marred or damaged by the use of a chisel to tighten or loosen.

In case the cylinder head joint leaks, if the nuts on the studs across the head from the leak be slacked off and the nuts immediately adjacent to the leak be tightened up, followed by the tightening of the nuts loosened, in many cases the leak will stop.

In replacing cylinder head gaskets or valve cages the surfaces should be thoroughly cleansed. Small carbon particles or grit often cause the gaskets to leak.

To detect whether the several valve rocker arms have clearance between the rollers and the cams, the engineer should spin the roller; the spinning of the roller denotes the existence of a clearance. This scheme is used at times to determine the point of opening the fuel valve. When the roller touches the cam nose it is no longer possible to spin it, and the engineer then assumes that the valve is just opening. Due to wear in fulcrum pin and roller pin, this lack of spinning may not actually prove the opening point. The best way is to listen for the sound of the escape of air as the valve opens.

A typical valve setting for a Diesel engine is as follows:

Fuel valve opens before top dead center....................	6 deg.
Fuel valve closes after top dead center......................	36 deg.
Exhaust valve opens before bottom dead center.............	42 deg.
Exhaust valve closes after top dead center..................	16 deg.
Admission valve opens before top dead center...............	24 deg.
Admission valve closes after bottom dead center.............	30 deg.

Each design of engine has a different timing, especially of the admission valve opening and closing points.

Two general designs of fuel or spray valves are in use: (1) The *open type;* (2) the *inclosed type.* With the latter, such as is used on the Snow and Allis-Chalmers engines, the oil is deposited in a receptacle opening into the combustion chamber. The spray valve holds back the injection air, which, as soon as the spray valve opens, blows through the receptacle, carrying the oil along with the air into the cylinder. In the *closed type* the spray or fuel valve is placed *between* the oil receptacle and the combustion chamber, with the air resting above the oil. When the spray valve opens, the air blows the oil into the cylinder.

The fuel valve casting is always water cooled. If not, the heat from the engine cylinder would raise the temperature of the oil to the ignition point; an explosion might then take place in the valve cage—this has happened when using light oil.

The thermal efficiency of the Diesel engine may be found from the indicator card by the following formula:

$$E=1-\frac{1}{n}\left(\frac{Td-Ta}{Tc-Tb}\right)$$

Where $E =$ efficiency;

$N =$ ratio of specific heats or $\frac{Cp}{Cv}$ usually taken for oil engine as 1.35;

$Td =$ temperature (absolute) at point of exhaust valve opening;

$Ta =$ temperature (absolute) of suction air usually taken as 673;

$Tc =$ temperature at end of combustion;

$Tb =$ temperature at end of compression.

These temperatures can be calculated from the indicator diagram as follows: Assuming Ta, the temperature of the suction air to be 673, and the pressure at this point the value shown by the indicator card and calculating Va, the cylinder volume at this point from the known stroke, bore and percentage of combustion space, then

$Pa\ Va = M\ R\ T$ *or* $K\ T$ from the gas formula $P\ V = M\ R\ T$; the value of K can be determined then by substituting the values of Pa, Va, and T. Using this same value of K,

$$\frac{Pe\ Vb = K\ Tb}{Pd\ Vd = K\ Td} \text{ and}$$

all necessary values of temperature can be obtained by measuring the pressure and volume from the indicator diagram and inserting in the formula. Substituting these values of T in the efficiency formula, the theoretical thermal efficiency may be obtained. This value is comparative only since a certain weight of fuel and injecttion air has been added which will alter the actual values.

From the efficiency formula

$$E = 1 - \frac{1}{n}\left(\frac{Td - Ta}{Tc - Tb}\right)$$

it is evident that the efficiency increases as the period of fuel valve opening is shortened. For, with a given weight of oil to be injected, the temperature of the cylinder contents increase as the period of burning decreases. Td then increases if the period of injection is decreased.

The fraction $\frac{Td - Ta}{Tc - Tb}$ becomes less since the denominator has increased.

The actual fuel consumption per brake horsepower hour of a Diesel engine ranges around 0.45 lb. in usual sizes. In engines of 1,000 hp. and over a consumption as low as 0.38 lb. of fuel has been obtained. A horsepower hour theoretically requires 2,545 B.t.u., which, with oil having 18,500 B.t.u. per pound, is approximately .14 lb. of oil. The actual Diesel requires 0.42 lb.; in other words, the Diesel's thermal efficiency based on brake horsepower is $\frac{.14}{.42}$ or 33⅓ per cent of the theoretical efficiency. Since the mechanical efficiency is around 75 per cent, the thermal efficiency based on indicated horsepower is 44 per cent. The ideal engine working between the temperatures of the Diesel engine would have an efficiency of about 57 per cent; therefore, it is seen that the actual engine obtains 33/57 or 58 per cent of the perfect engine. This is in striking contrast with the steam engine which secures about 10 per cent of the possible heat in the steam supplied to the engine.

The usual commercial Diesel carries a mean effective pressure at full load, based on brake horsepower, of 88 to 96. Since the indicated

efficiency decreases with an increased M. E. P., while the mechanical increased with the M. E. P., there is a mean value of the latter, at which the greatest overall efficiency is obtained.

Indicator diagrams should be taken at frequent intervals. They are valuable in determining the action of the working cylinder. The indicator and rigging must be in good condition or baffling diagrams will be taken. Diagrams taken with the rigging connected at right angles to the crank pin are best to determine the condition during the period of combustion.

In starting a Diesel the cylinders should not be overprimed; if the starting cylinders get much fuel, there may occur a preignition resulting in a bent connecting-rod or fractured cylinder head.

The starting cylinder should not be turned over too many times under air pressure. The air, as it expands through the valve, chills the entire cylinder. Frequently the cylinder becomes so saturated with moisture that it will not fire when the fuel is turned on.

Some Diesels have four lubricating oil feeds to each cylinder; others have three and some but two feeds. In practice two feeds, if on opposite sides of the cylinder, are quite effective. The oil should be introduced when the piston is on the suction stroke. The piston then smears the oil on the entire cylinder surface on the compression stroke. A rate of 20 drops per minute per feed in case of two feeds per cylinder or 10 drops for a three or four feed system is ample. In inclosed frame engines with pressure oiling systems the lubrication to the piston can be reduced, since the piston picks up a great deal from the crank case.

The piston pin requires much attention as to its lubrication. It is located in a belt of high heat and, if oiling is stopped, the pin will seize the brass.

Many small carbon particles drop alongside the piston and get into the oiling system. A filter alone will not eliminate all this carbon, and a filter press will do much toward clarifying the oil in the lubrication system. The filters and filter presses should be cleaned at least once in two weeks, at which time a small amount of make-up oil should be added to the system.

A Diesel should be able to deliver from 6,000 to 10,000 horsepower hours per gallon of lubricating oil—in most cases the engineer obtains not over 3,000 horsepower hours.

A typical lubrication oil specification is as follows:

Boiling Point..600—700° Fahr.
Flash Point..325—500° Fahr.

Fire Point	400—600° Fahr.
Viscosity at 100° Sayboldt	550—800
Specific Gravity, Baumé	18—24
Carbon Content, per cent	0.05—0.2
Sulphur	None

While an asphaltum base lubricant appears to give as good results as does a paraffin base oil, many engineers prefer the latter. To determine the base, take a piece of paper and drop a little oil on it. If, upon evaporating, the paper returns to its original color, the oil has a paraffin base. If the oil has an asphaltum base, the paper will have a darker color.

To determine whether the oil contains much gummy matter, place a little on a piece of glass; the residue left on evaporation will reveal the amount of gummy matter carried.

If the oil carries any sulphur, a piece of bright brass will become tarnished after being suspended in the oil for thirty-six hours.

Every Diesel engine station should have a small storage tank holding a day's supply of fuel oil. Accurate record should be kept of the fuel consumption per hour. Any misadjustment of the engine quickly shows up in the oil consumption.

The engineer should learn to detect a warm bearing by the smell of the crank case; no better guide to the engine's condition exists than the smell.

The plant storage tank should be located several feet, 25 feet if possible, from the engine room to avoid fire risks. If the oil is very heavy and does not pump easily, a hot water pipe coil should be placed in the storage tank. Part of the jacket cooling water may be bypassed and sent through a coil in or around the exhaust pot. This water will be hot enough to cause the oil to flow freely.

The amount of cooling water required by a Diesel engine varies with the temperature of the water. Theoretically the amount

$$W = \frac{XH}{100\ (T_1 - T_2)}$$

When

X = percent of heat in cylinder absorbed by the water;
H = heat of the oil to the engine;
T_1 = discharge water temperature;
T_2 = inlet water temperature;
W = weight of water per hp. hour.

Since usually $\frac{XH}{100}$ approximately equals to 3000 B.t.u. per horsepower

hour, the formula can be written

$$W = \frac{3000}{T_1 - T_2}$$

In case of a 500 horsepower engine

$$W = \frac{500 \times 3000}{80-120} = 3750$$

or 75 lb. per horsepower hour. Usually the cooling water can be secured at a temperature below 80 degrees. Eight gallons per brake horsepower hour is generally taken as an average value.

Two general types of cooling-water circulating systems are used: (a) The closed system where the water jackets are under the pressure due to the discharge of the water over a cooling tower; (b) the open system wherein the discharge is open, the water flowing into a sump and then being forced by a pump over the cooling tower. The latter is preferable as the operator can see the circulation of the water.

In localities where the water carries much minerals, etc., it is advisable to use a closed system filled with distilled water. The water is passed through a pipe coil similar to an ammonia condenser. A supply of cold water dripping on the outside of the pipe cools the circulating water. The raw water is then cooled in a tower or by evaporation as it drips from pipe to pipe. An exhaust distiller furnishes any distilled make-up water made necessary by leakage, etc.

Many plants have installed water-treating systems, in which soda ash and other chemicals are used to purify the water.

An overhead storage cooling water tank should be installed at all, save very small plants. The tank should hold at least one hour's supply.

Where a large amount of fuel oil is carried in storage, a concrete tank is far cheaper than a steel one. It should be well reinforced with steel bars and wire netting. A square tank is easier to make than a round one but is more likely to crack.

While the majority of Diesel stations use a so-called Diesel fuel oil, of about 28—32 gravity Baumé, this oil costs more than a heavier oil. If the plant is large, it will pay to use the cheaper grade oil. A number of Diesels are burning Mexican crude oil very successfully. The heavy oils require a more thorough filtering, since much sediment is carried in suspension. A filter carrying either a felt or terry cloth filtering basket should be used.

Where a heavy oil is burned, it is expedient to install a small tank and use kerosene to start the engine. Kerosene should also be used for

a few minutes before stopping; this will clean the fuel valve and combustion chamber. With heavy oils a higher compression pressure is usually necessary.

Exhaust pipes should always be water-jacketed. A number of fires have occurred by oil leaking out of the fuel lines and striking a red-hot exhaust pipe. It is never advisable to use a water jet inside the exhaust pipe. If sulphur is present it will eat the iron pipe.

Two of the best indications of the way the engine is working are the sound and color of the exhaust gases. Most engines have test plug holes in the exhaust elbows.

If a cylinder gives out a sharp metallic sound, it is evident that the fuel is being admitted too early. If a dull pound is heard, the fuel admission may be late or a crank pin bearing may be very slack.

A sharp pinging or knock (known as "pinking") is not objectionable. It has been proven by many experiments that this detonation or "pinking" does no harm; in fact, the engine is working most efficiently at this time. If the pound is very violent, of course, there is danger of a bent connecting-rod. Increasing the rocker roller clearance will make the fuel valve open later, and this will eliminate the knock.

To determine the proper degree of clearance in the piston pin bearing, suspend the piston and rod from a chain hoist. If a man can just swing the rod without tilting the piston as it hangs free, the play is ample. If the rod swings freely, there is too much play.

It never pays to slack off the piston pin bearing nuts to give more clearance without inserting enough shims between the two halves. If the shims are not inserted, the brasses have a slight play and will ultimately batter.

On account of the relatively short connecting-rod and the high pressure common to Diesel engines, the resultant side pressure on the cylinder wall is high. Therefore, the trunk pistons must be correspondingly long to keep the unit pressure within safe limits and to reduce excessive cylinder and piston wear. For this reason cross-heads and guides were used during the earlier development of Diesel engines in Europe. On account of its many advantages the trunk piston has been adopted and is used in engines, even of 250 horsepower per cylinder and in units of 1,000 horsepower. The trunk piston is more easily provided with a greater surface than a crosshead, thus insuring less wear; the lubrication under pressure of a cylindrical guiding surface is more effective than with open guides. The piston moves over perfectly cooled walls, whereas the crosshead tends to heat more readily and when once

hot is not easily cooled; moreover, water cooled crossheads complicate construction. The use of crossheads and guides greatly augments the height or the length of the engine, increasing its cost. For large engines this construction is used, as it affords accessibility and ease of adjustment, the crosshead guide in these engines being watercooled.

Flywheel construction does not differ from that common to gas-engine practice. To insure a fairly uniform speed, a flywheel becomes necessary. On account of the high compression and the mean effective pressures, the cranks are subjected to severe duty and transmit correspondingly high efforts. During the power impulse, excess power must be stored in the flywheel to be released during the compression stroke and during the other two unproductive strokes in engines with a four-stroke cycle. One-cylinder engines, especially if great uniformity of speed is demanded as in driving two or more alternators in parallel, require an excessively heavy flywheel. Concentraton of all the power in one cylinder also greatly increases the comparative weight of the engine.

Next to the fuel pump, the fuel-injection valve is the most important and characteristic part of a Diesel engine, and on its correct construction greatly depends the satisfactory operation of the engine. It performs two distinct functions: First, it has to admit the liquid fuel into the engine cylinder at the proper time and must therefore be an accurately timed valve. Second, it has to introduce the liquid fuel into the cylinder in a completely atomized form and must therefore be an efficient atomizer. The first requirement is fulfilled by storing around the tip of the fuel needle a drop of oil, which is forced ahead of the injection air into the heated atmosphere of the engine cylinder at regularly recurring intervals and thus initiates the ignition. The second requirement is accomplished by dividing the fuel supply into numerous small fuel particles in the fuel valve, preceding atomization and injection.

To seal the air chamber of the fuel valve and prevent leakage of the air around the valve needle, lead or Babbitt-metal shavings mixed with flaked graphite are used as packing material, secured by appropriate glands.

From five to seven snap rings are used to seal the cylinder. To hold the rings closed they are pinned together at the lap joints. They are pinned to the piston by a small dowel set in a hole in the grooves. Such fastening is always necessary in two-cycle engines to prevent the ends of the snap rings from shifting around the piston and thus traveling across the exhaust ports and catching in them while they are being uncovered or covered by the piston.

The reliability of the fuel pump and its sensitiveness in reponding to the slightest variations in load largely determine the satisfactory

operation of the engine. The pump must respond to the governor instantly and must furnish a supply of fuel accurately proportioned to the needs of the engine load. The permissible variations have to deal with extremely small volumes of fuel. A fuel volume of 0.15 cu. inch corresponds to a full-load injection for a 50-horsepower engine with about 80 injections per minute (160 revolutions per minute). A variation in load of one percent amounts therefore to a fuel variation of only 0.0015 cu. inch and illustrates how seriously the regulation of an engine may be affected by minute quantities of fuel, unless the fuel pump and governor measure with great precision the fuel required.

In pumps that are driven by the vertical governor shaft, which, in an engine having a four-stroke cycle, makes double the number of revolutions of the camshaft, the plunger delivers the fuel in two parts for any one fuel charge, corresponding to one revolution of the camshaft. The governor, by acting between the two plunger strokes, can therefore adjust the second part of the fuel charge to the new engine load. As a result of the desire to improve the regulation further, individual pumps are often built as muti-plunger pumps, one plunger being provided for each fuel valve (each cylinder). As the plungers are driven simultaneously by one eccentric device, they divide the fuel effectively, but the governor continues to influence only the combined quantitiy of fuel delivered by all the plungers and not of that delivered by each plunger successively.

The exhaust pipes should be of cast iron as steel pipes corrode rapidly, especially when the fuel oil contains an appreciable quantity of sulphur. The sulphur burns to sulphur dioxide, which may be oxidized to the trioxide in the engine cylinder and combine with the water vapor of combustion to form sulphurous or sulphuric acid. The exhaust gases should never be cooled to the condensing point of water, which would cause corrosion. As the exhaust gases issuing from the engine are hot (600° to 1,000° F.), the exhaust pipes are jacketed and water-cooled in the proximity of the engine to make work around the engine bearable to the attendants and to prevent burns.

When the engine is stopped, the water should be continued in circulation for some time, as the heat stored in the piston, the cylinders and the cylinder head is considerable, being sufficient to bring the water to boiling point so that sudden stopping of water might cause the cracking of a cylinder or of the cylinder head.

Diesel engines may be classed as low-speed and high-speed engines. The former are preferred for hard, continuous duty. They have relatively low piston speed, varying from 600 to 800 feet per minute, the piston speed increasing with the power. The number of revolutions per minute varies from 250 to 150, decreasing with the size of the engine. The

stroke-bore ratio varies from 1.3 to 1.9, the higher ratio being preferred for low-speed engines for hard service, although with an increase in piston speed the stroke-bore ratio decreases, likewise the number of revolutions. High-speed engines have a piston speed of 700 to 1,000 feet per minute, a stroke-bore ratio of 1.0 to 1.3, and an engine speed of 250 to 350 revolutions per minute. The speed of engines for special purposes, as for submarines, is often increased to 500 and 600 revolutions per minute with low stroke-bore ratio to obtain a light engine of low height.

At higher altitudes the specific duty of Diesel engines decreases appreciably, owing to the lessened density of the air, which affects the engine just as a low volumetric efficiency would. The horsepower rating of the engine decreases 3 per cent for every 1,000 feet of added altitude. Nearly 40 per cent of the power loss could be recovered by precompressing the rarified air to atmospheric or slightly higher pressure in positive-pressure blowers and filling the engine cylinder with this air. Blower equipment is considerably cheaper per horsepower of capacity than Diesel engine equipment. At high altitudes it may pay, under certain conditions, to install blower equipment for precompressing the air for engines having a four-stroke cycle. An engine having a two-stroke cycle can compress the air readily by using a larger scavenging pump.

A desirable petroleum fuel for Diesel engines should have the following properties:

(a) It should burn completely without leaving any residual matter in the cylinder, either in the form of soot, coke, or ash.

(b)) It should be free from mechanical impurities which might clog the fuel pipes, the valves of the fuel pump, and the fine fuel passages in the fuel-injection valves and nozzle, or might cause excessive cylinder wear.

(c) It should be sufficiently fluid at ordinary temperatures to flow readily to the fuel pump and thence to the fuel-injection valve.

(d) It should be free from water, as water lowers the heating value of the oil and may prevent its ignition.

(e) It should be free from highly volatile oils, which will evaporate at ordinary temperatures and form an inflammable mixture with the air, thus introducing a fire hazard.

(f) It should have a high heating value.

As a general rule the smaller the proportion of residue remaining at a temperature higher than 400° C., and the greater the volume of vapor coming over between 200° and 400° C., the better is the oil suited for use in Diesel engines. A further valuable criterion of the burning qualities of oils that leave residues or "oil tars" at temperatures higher than 400° C. is the quantity of coke left on distilling the residue at temperatures higher than 400° C. The greater the quantity of

coke the less suitable is the fuel. Fuel oils with a coke content can not be used in all engines, and a content of 5 per cent may be taken as the upper limit for all engines.

Ash is the most detrimental remnant of the burning of fuel oils for Diesel engines, as it causes excessive wear of cylinders and exhaust valves. The ash is usually composed of mineral particles of great hardness, such as quartz and silicates, or oxides of iron and aluminum, which becoming mixed with the film of lubricating oil, adhere to the piston and the cylinder walls, accumulate, and cause excessive wear. An ash content in excess of 0.05 per cent will render an otherwise excellent fuel unsuitable for use in a Diesel engine.

Sulphur in oil burned in the engine cylinder is converted to sulphur dioxide and may be oxidized to the trioxide. At the high temperature prevailing the percentage of sulphur usually present in fuels has no appreciable effect on cast iron. Even so high a sulphur content as 5 per cent in the fuel corresponds to less than 0.1 per cent by volume of sulphur dioxide in the gaseous products of combustion, which are replaced by a fresh volume of air every revolution in a two-stroke engine. Mexican oils containing sulphur up to 5 per cent are successfully burned in Diesel engines; the exhaust pipes, however, should be of cast iron rather than steel, and the cooling of the gases should not be carried for far that the water vapor resulting from the combustion of the hydrogen will condense, as then the corrosive action of the sulphurous and sulphuric acids would be more destructive.

Certain important adjustments are made for burning heavy oils, as follows: Increased injection-air pressure and smaller nozzle orifice to insure more complete atomization of the fuel, higher compression in the engine itself to procure high temperature of the air of combustion; and heating of the oil to as much as 180° F. to make it more fluid and more easily atomized. These different adjustments require considerable experience on the part of the operating engineer, who must take into account the character of the fuel used.

The following specifications are quite broad and cover oils that can be successfully burned in the Diesel engine. Some plants use oils of a heavier character where the cost of the lighter oils is such that the lower price of the heavier oils compensate for the extra difficulties.

Fuel Oil Specifications

Heat value.—Not less than 18,500 B.t.u. per pound. Contractor must give low heating value of the fuel supplied.

Gravity at 60° F.—With engines having closed fuel nozzles the oil shall not be heavier than 20° Baumé. For engines having open

nozzles the oil should not be heavier than 16° Baumé. The oil should not be lighter than 36° Baumé.

Residue.—Not more than 10 per cent. The residue is that portion of the fuel remaining in the cup after being subjected to a temperature of 300° C. (572° F.) for 120 hours.

Flash Point.—From 125° to 150° F.—Dependent on the engine.

Burning Point.—From 160° F. to 300° F.

Sulphur.—Not over 2.0 per cent.

Water.—Not over 0.3 per cent.

Ash.—Not over 0.05 per cent.

Gravity.—The usual method of indicating the weight of crude or fuel oil is by the Baumé scale. In the table below is given the Baumé scale with the corresponding gallon weight and specific gravity.

Baumé Degree	Pds. Per Gal.	Spec. Gravity	Baumé Degree	Pds. Per Gal.	Spec. Gravity
10	8 336	1 000	25	7 527	0 903
11	8 277	0 993	26	7 477	0 897
12	8 219	0 986	27	7 435	0 892
13	8 161	0 979	28	7 385	0 886
14	8 102	0 972	29	7 344	0 881
15	8 052	0 966	30	7 294	0 875
16	7 994	0 959	31	7 252	0 870
17	7 935	0 952	32	7 202	0 864
18	7 885	0 946	33	7 160	0 859
19	7 835	0 940	34	7 119	0 854
20	7 777	0 933	35	7 069	0 848
21	7 727	0 927	36	7 027	0 843
22	7 677	0 921	37	6 985	0 838
23	7 627	0 915	38	6 944	0 833
24	7 577	0 909	39	6 902	0 828

The conversion of chemical energy into mechanical energy is accomplished by means of an engine, steam or internal combustion. If a steam engine is used, an intermediate member is placed in the chain, this member being the boiler. The chemical energy of the fuel is changed into heat energy in the furnace and this heat energy is transferred to the water in the boiler. The water in this case acts as a carrier and does not enter vitally into the whole problem of heat conversion by means of the steam engine. That is to say, other carriers could be used instead of water, but it happens that water is plentiful and has certain characteristics which make its use as a carrier desirable in many ways. In the engine cylinder the heat in the steam is converted into mechanical energy, by moving a piston to and fro in this cylinder. In some cases the reciprocating motion is used as such; in the great majority of engines, however, the reciprocating motion is changed into a rotary motion by means of a system of members forming a kine-

matic chain. In the Diesel engine combustion takes place directly in the cylinder, that is, the chemical energy is changed into heat energy in the cylinder, which acts, in this case, as the furnace of the boiler. The medium or carrier in the Diesel engine is formed from the fuel and the air required for combustion. This acts very nearly as a perfect gas, while the action of steam in a steam-engine is far from that of a perfect gas.

If the actions of a steam engine and a Diesel engine are almost identical, why use one in preference to the other? The answer, theoretically, is this: The efficiency of the Carnot cycle, the highest attainable, is $\frac{T_1-T_2}{T_1}$ where T_1 is the absolute temperature of the high point of the cycle and T_2 is the temperature of the low point, or, in other words, the cycle works between limits of T_1 and T_2. The value of T_2 is fixed by the atmospheric temperature and is the same for both the Diesel engine and the steam engine, assuming both these engines to operate on the Carnot cycle. In the steam engine, the value T_1 is about 862 degrees for 250 pounds steam pressure. With a value of T_2 at 70° F. or 530 degrees absolute the efficiency would be 0.385. With a temperature of 2,500° F. in the cylinder of the Diesel the efficiency would be 0.83 or more than twice that of a steam engine. Possibly, then, the Diesel engine has the advantage of the steam engine in the ratio of 0.83 to 0.385; but theoretical considerations alter this considerably since neither the steam engine nor the Diesel attains these temperature ranges.

In large Diesel engines using piston cooling the joints of the water pipes must be kept watertight. If the water drips into the crank case and mixes with the lubricating oil there is a dangerous condition created. The water and oil will form an emulsion which will clog up the oil pipes. If carbon particles drop from the piston into the crank case the tendency of the oil and water to emulsify is increased and the ability of the oil to settle out of the water is reduced. Any foreign matter acts this way; rust especially is quite active. If water does leak into the engine frame the mixture should be drawn off and allowed to separate.

A frequent source of trouble in the early Diesels was the "sticking" of pistons. This was generally due to improper lubrication. If an excess of lubricating oil is fed to the piston the excess will not be burned clean; instead, a gummy residue will be left on the piston and cylinder walls. As long as the engine is working this tar will remain hot and yielding. When the engine is stopped the tar cools off into a hard mass which grips the piston and cylinder, holding the two in a vise-like grip. On attempting to start the engine, the operator finds

the piston "frozen." It often required the use of a screw jack to break the tar bond. With a better knowledge of Diesel lubrication requirements prevailing, "frozen" pistons are no longer encountered.

Supplies of new lubricating oil should in no case be placed in tanks that have been used to store fuel oil without a thorough cleaning.

No man should ever be allowed to enter a fuel oil storage tank until the cover has been removed and the tank allowed to clear itself of gases. At least twelve hours should be taken for the airing process. A number of lives have been lost because of failure to observe such precautions. Likewise an open flame should never be allowed around an oil tank; no light save an electric bulb with a wire guard should ever be used. If the bulb is unguarded it may break and the heat of the filament has ignited gases in a number of instances.

A sudden rise in pressure of a lubrication system is usually proof of a stoppage of one or more of the oil pipe lines.

In marine Diesels there is more or less danger of salt water getting into the oiling system, especially if the pistons are salt water-cooled. Frequent inspection should be made; a silver nitre test will reveal any contamination of the oil by the salt water.

In a Diesel driven ice plant, a ton of ice can be produced at the expenditure of 72 horsepower-hours, or with a fuel consumption of approximately 4 gallons per ton of ice. With fuel oil at ten cents the power cost will be 40 cents per ton. To equal this electricity must be purchased for less than 1 cent per kilowatt.

A Diesel driven flour mill produce a barrel of flour at an expenditure of 0.55 gal. of fuel oil, 0.006 gal. of lubricating oil, and with 1 cent of repairs. The total power cost including fuel, repairs, wages, etc., is 9 cents per barrel.

A Diesel cement plant will turn out a barrel of cement at a power cost of 10 cents; this does not include interest charges.

A Diesel central station will put a kilowatt-hour upon the switchboard at a station cost of less than 5 mills; the total cost including overhead charges is less than 9 mills per kilowatt-hour.

In a series of tests by the British Admiralty it was found that the solid-injection or pump-injection Diesel gave the same economy as did the air-injection Diesel. With the engine under test the maximum efficiency was 0.425 lb. of fuel per brake-horsepower-hour. It was found that the solid-injection engine was able to operate at 20 r.p.m. while the

air-injection engine began to misfire at a much higher speed. This was due to the chilling action of air blast.

Experiments in England have revealed that air-injection Diesels can develop more power per unit cylinder volume than can the solid-injection Diesel. It was also found that above a certain air pressure, depending upon the engine design, the engine power dropped due to excessive chilling of the combustion space.

The air intake pipe should be covered with a cloth screen to catch all dust particles. If possible the air line should be run outside the engine room. Fresh air is secured in this way and danger of drawing oil-charged air into the working cylinders is entirely eliminated. For the same reason the intake to the air compressor should be from without the engine room.

In some two-cycle Diesels, to secure complete scavenging of the exhaust gases, a second row of air ports is placed in the cylinder walls. These are controlled by a rotary valve and are opened after the exhaust ports are uncovered, and stay open after the exhaust ports are covered. This supplies a cylinder full of pure air.

The largest American Diesel has a capacity of 400 horsepower per cylinder, the stroke being 46 inches. The smallest American Diesel offered for commercial use is 50 horsepower.

The total horsepower of "true" Diesels, i. e., Diesels using air-injection and having a compression pressure of at least 450 lb. per sq. inch is 260,000. There are 17 firms engaged in the manufacture of Diesel engines in the United States.

The total output of Diesel engines in the United States in 1921 was 103.

In figuring the overhead charges of a Diesel station the only unknown value is the percentage to be charged as a depreciation. Some engineers figure as high as 8 per cent, thinking the life of the Diesel is 10 years. On the contrary there is every reason to believe that 5 per cent depreciation is ample. There are at least twenty-five Diesels in the U. S. that have been in operation since 1903-1905, and are in fair condition. The life of a Diesel is at least 20 years.

Unlike the steam engine or steam turbine which show a large increase in fuel per brake-horsepower as the load drops, the Diesel has an excellent low load efficiency. For this reason the Diesel is economical on changing loads.

CHAPTER XVII

Diesel Engines for Railroad Service

A Complete Description of Present Developments—The McIntosh and Seymour Railroad Cars—Description of the Engines—Transmission—Control—Results as Regards Maintenance and Upkeep—Comparative Figures of Operation of Steam and Diesel—Ingersol-Rand Locomotive—Particulars of Equipment—Details of Engine—Baldwin-Knudsen Locomotive—Description of the Engine—Dual Crankshafts—Port Scavenging—Special Pistons—Fuel Pumps—Cooling Systems—Control Gear.

The advantages of the Diesel engine for railroad use compared with the orthodox steam locomotive are many. The modern high pressure steam locomotive with all its up-to-date improvements only has a degree of efficiency of 7 per cent, while that of the Diesel locomotive attains an efficiency of 33 per cent. Other advantages are,—it is ready for instant service; it makes possible the elimination of coaling plants, ash pits, turn tables, roundhouses and hostling services—all of which are required for steam locomotives. Very little water is necessary, thus eliminating costly watering stations and troubles due to bad water conditions. Its availability for service is approximately 80 per cent, or double that of the steam locomotive, and its cost of maintenance is less than of the other. Because of its smaller mechanical parts it eliminates the necessity for heavy shop machinery.

The Diesel-electric locomotive provides uniform continuous torque at the wheel rims which results in less wear on the tracks and provides higher tractive effort at starting and slow speeds on lighter axle loads. The Diesel engine can be loaded full at all speeds which aids in securing fuel economy.

The electric transmission has an advantage over gear transmission in that it is self-shifting. The speed ratio between the generator and the driving motors automatically adjusts itself to meet the demand of the service. This type of

transmission also protects the oil engine from the strains and shocks to which it is subjected with a mechanical driving transmission between engine and driving wheel.

There is no fixed relation between the engine speed and the locomotive speed; thus the engine can always be run at its most economical speed, whether the locomotive is running

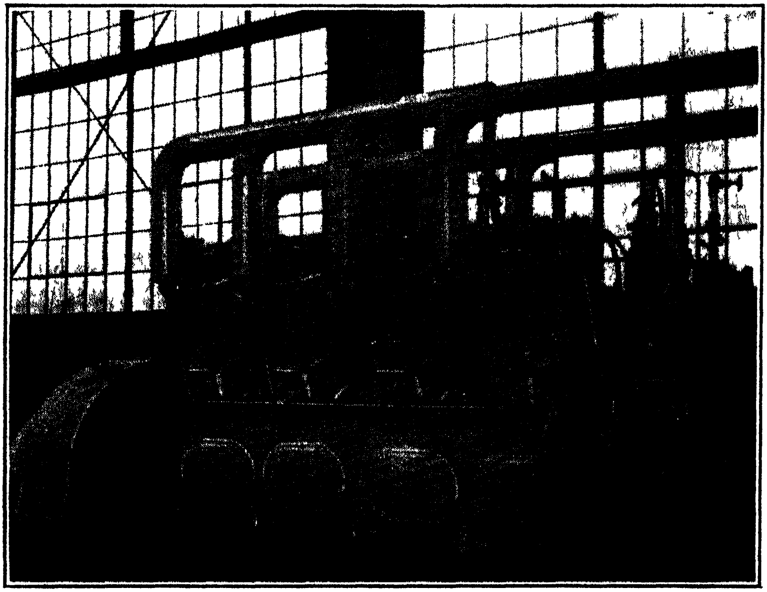

Fig. 320.—200 B.H.P. McIntosh and Seymour 8-Cylinder V-Type Diesel Engine for Railway Use

at a high speed on level track or at a low speed during acceleration or when hauling a heavy train. The electric transmission is equivalent to a gearing with an infinite number of gear changes.

Very little has been accomplished in America in connection with Diesel engines for railroad use, but it is a field in which several of the largest concerns appear to be considering much more favorably than in the past.

McIntosh and Seymour Engines. McIntosh and Seymour Diesel Engine Co., of Auburn, N. Y., are developing the Diesel

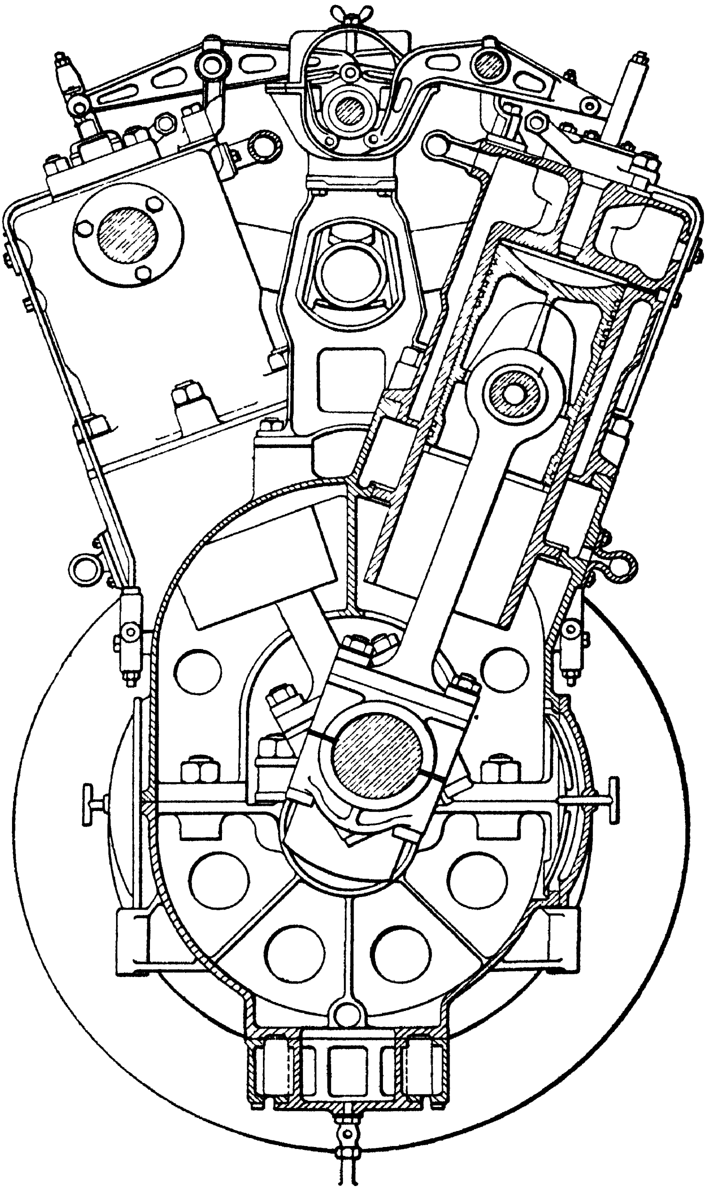

Fig. 321.—Sectional View of McIntosh and Seymour Engine

locomotive and are associated with the Swedish firm, The Aktiebolaget Atlas Diesel Engine Co.

Four types of cars have been developed and fitted with 75, 120, 160 and 250 B.H.P. engines of the four-cycle type. The 75 H.P. unit is built with six cylinders in line and is operated at a speed of 550 R.P.M.

Fig. 320 shows an eight cylinder, 8 inch bore by 12 inch stroke engine built at the Auburn shops. This engine develops 200 B.H.P. and runs at 500 R.P.M. and was developed especially for railway use.

A cross-sectional drawing of this engine is shown in Fig. 321. It will be noticed that the crankshaft is entirely enclosed in an oil tight crank case upon which the two rows of cylinders are mounted, being placed at an angle of 40 degrees.

A general view of these engines are shown in Figs. 322, 323 and 324. The base plate carries the main bearings which are removable and lined with white metal in the usual manner. The crank case is mounted upon this base and is provided on either side with large and easily removable covers so as to give the greatest possible accessibility to the internal working parts of the engine.

The cylinders are cast in one piece with the cylinder head, so as to eliminate any packing between the head and the cylinders and the valves in the head are fitted into castings that are easily removable for inspection or other purposes as required. Each cylinder is provided with one air intake valve, one exhaust valve and one fuel valve. An air starting valve is not ordinarily provided but arrangements are made for fitting this should circumstances merit the same, the motors usually being started by electric current to the generator from a storage battery. The operation of these valves is through the usual method of the cam shaft, which is placed between the cylinders at the upper end and is driven by means of bevel gears and vertical intermediate shaft from the crankshaft.

The pistons work in the customary manner with the connecting rods for each pair of cylinders side by side. The two cylinders which work on the same crank are offset to simplify the construction and eliminate the otherwise necessity of forking one of the connecting rods. This enables a lower end

bearing of the ordinary standard removable marine type to be used, which has been proved to be always reliable in service.

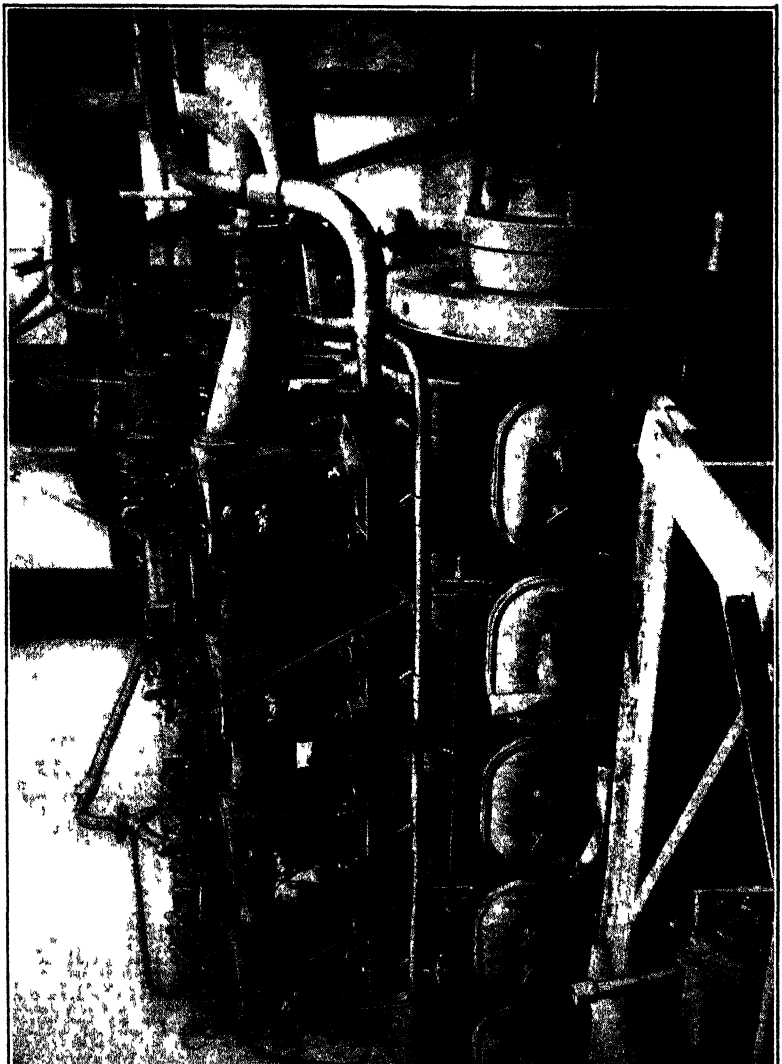

Fig. 322.—McIntosh and Seymour Railway Diesel Engine. Fly-wheel End

The compressor for providing the necessary air for fuel injection is mounted on one end of the crank case and is driven

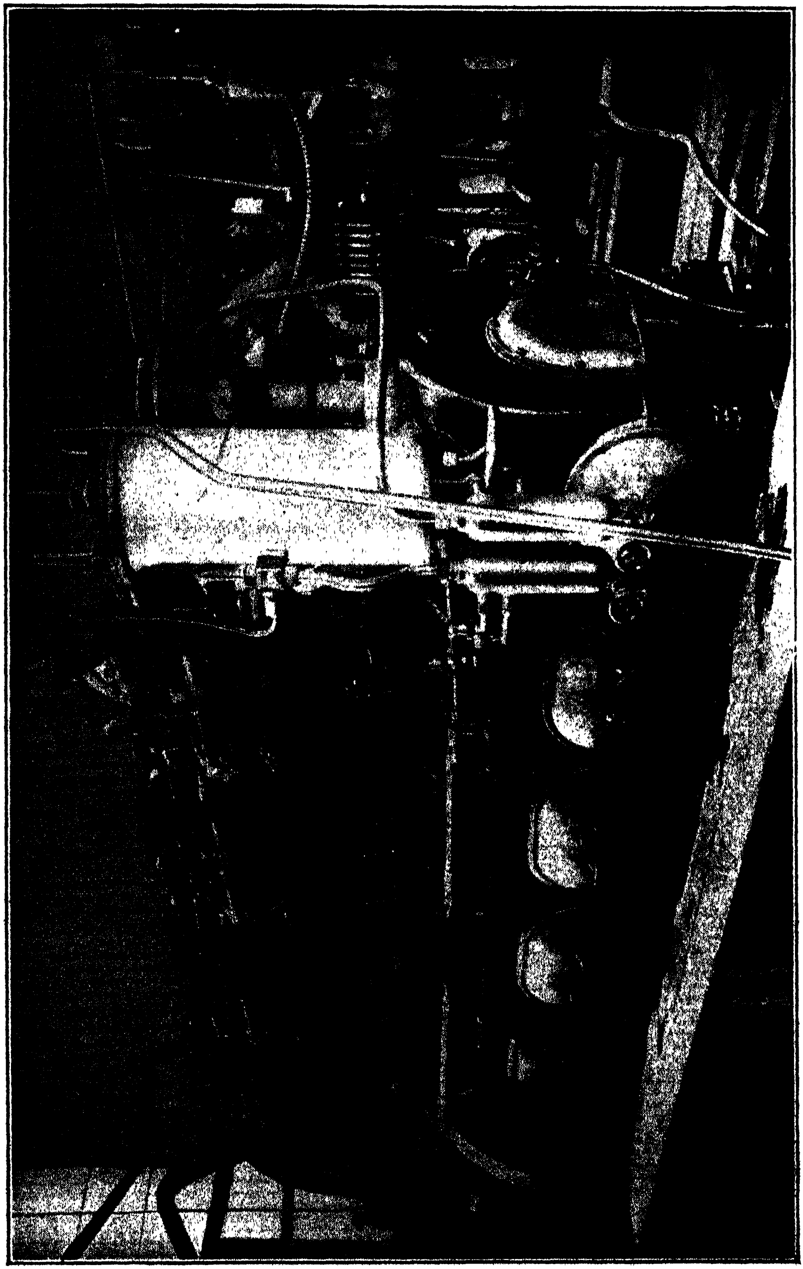

Fig. 323.—McIntosh and Seymour Railway Diesel Engine. Compressor End

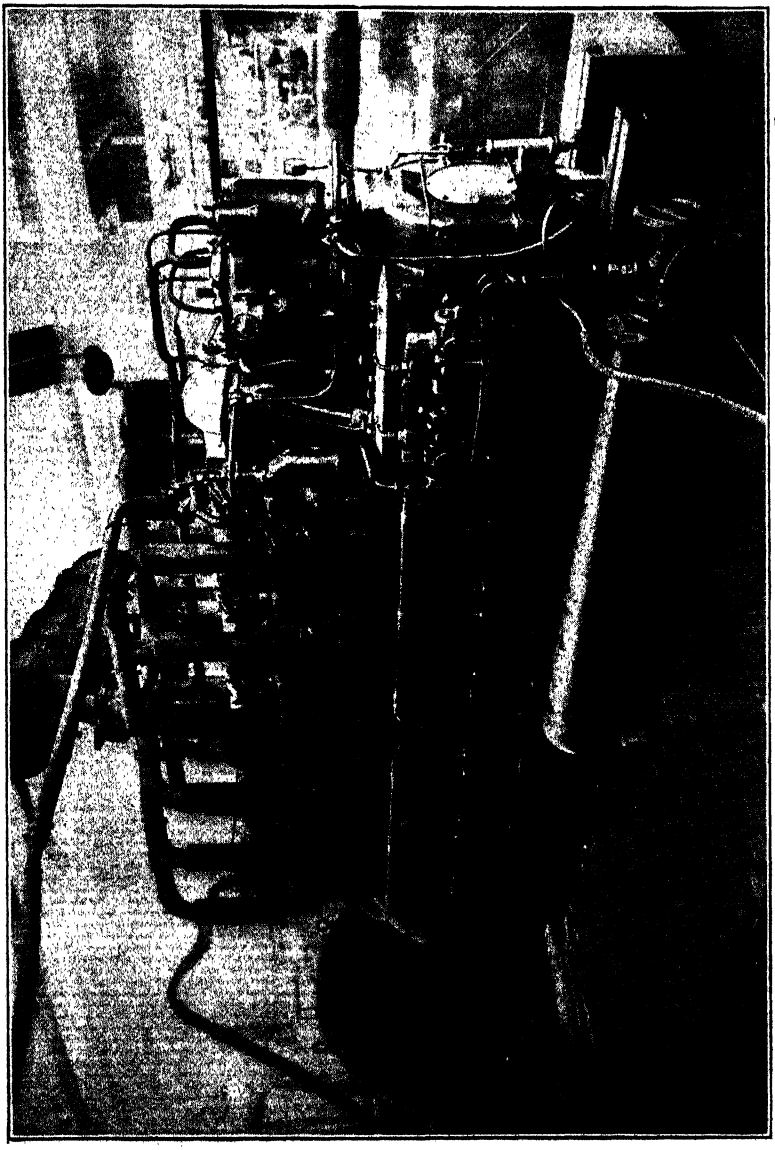

Fig. 324.—McIntosh and Seymour Railway Diesel Engine. Compressor End. Twelve Cylinder Type

by an extension of the crankshaft. This is a two stage compressor, the capacity of which is considerably reduced, owing to the fact that the only air necessary is that required for spraying the oil into the cylinders, the usual reserve of starting air being cut out, owing to the electrical system of starting mentioned above. When starting, however, the motor does not furnish enough air from the compressor to get under way and therefore the usual injection air bottle is fitted and enough air is kept in reserve for supplying the spray valves with the necessary amount of air until the motor is up to speed and the full quantity of air is supplied by the compressor. The flywheel coupling for the generator and the bevel gears for driving the cam shaft are all placed at the other end of the engine.

A single fuel oil pump supplies the fuel oil to all the cylinders, as in a small engine such as this such minute quantities of fuel oil are delivered for each power stroke that the single pump, together with the distribution system, is preferred. The fuel pump is controlled by the governor, which regulates the amount of oil supplied in accordance with the load, the oil being pumped to a distributor of the well-known Hesselman type. This distributor is so arranged that any cylinder can be cut out for temporary inspection if it is so desired. The fuel is injected at about 850 pounds' pressure and is sprayed by means of compressed air supplied from the compressor previously mentioned.

No special arrangements are provided for taking the air from the outside of the car, ordinary suction pipes being fitted so that the air is sucked from the inside of the car, a fresh supply being drawn in from the louvres in the sides and the roof of the car.

The cooling fan installed for the purpose of cooling the circulating water does the double duty of ventilating the car, as well as driving the air over the radiators for cooling purposes.

The cylinders and air compressor are water cooled, the water being kept in circulation by means of a pump driven from the cam shaft.

The hot water from the jackets is always utilized in the

winter time for heating the car; special regulating valves are provided in the engine room so that all or part of the water can be circulated around the car according to the requirements.

The main bearings are lubricated by pressure feed, and the oil is supplied in such quantity and under pressure that makes special provision for lubricating the cylinder unnecessary, the cylinders being supplied with a sufficiency of oil from that which is splashed around in the crank case from the main bearings and big ends. After being used in this manner, the oil is drained to the lower part of the crank case, from which it is pumped and passed through a filter and cooled in a cooler placed under the car, after which it is picked up by the oil pump and again circulated through the engine. The supply tanks for both fuel oil and cooling water are placed in the roof of the car, the tanks being filled by hand pump and hose connections from the engine room.

Below is a tabulation of resultant figures of a test made on one of these engines:

Cylinder diameter		$7\frac{7}{8}''$
Stroke		$9\,{}^{7}/_{16}''$
Revolutions per minute		500
Kind of fuel oil		Texas Oil
Test	No. 1	No. 2
Percent of load	100	100
Load on brake, lbs.	265	295
Length of brake arm, inches	$56\frac{3}{8}''$	$56\frac{3}{8}''$
Revolutions per minute	500	495
Brake horsepower	118	130
Length of test minutes	44	
Fuel oil consumption during test, lbs.		37.2
" " " per hour, lbs.		50.6
" " " " B.H.P. per hr., lbs.		.428
Injection pressure lbs. per sq. inch		795

Transmission. The electrical energy from the Diesel electric machinery is transmitted to the two driving axles and the arrangement is such that full effect of the Diesel engine can be utilized at all speeds of the car. Usually, the generator is

mounted on the same sub-base as the engine and is coupled to the latter by means of flexible coupling with the flywheel serving as a half of the coupling. The continuous current is regulated within certain limits up to 550 volts by means of magnetic control and the generator is arranged to serve for

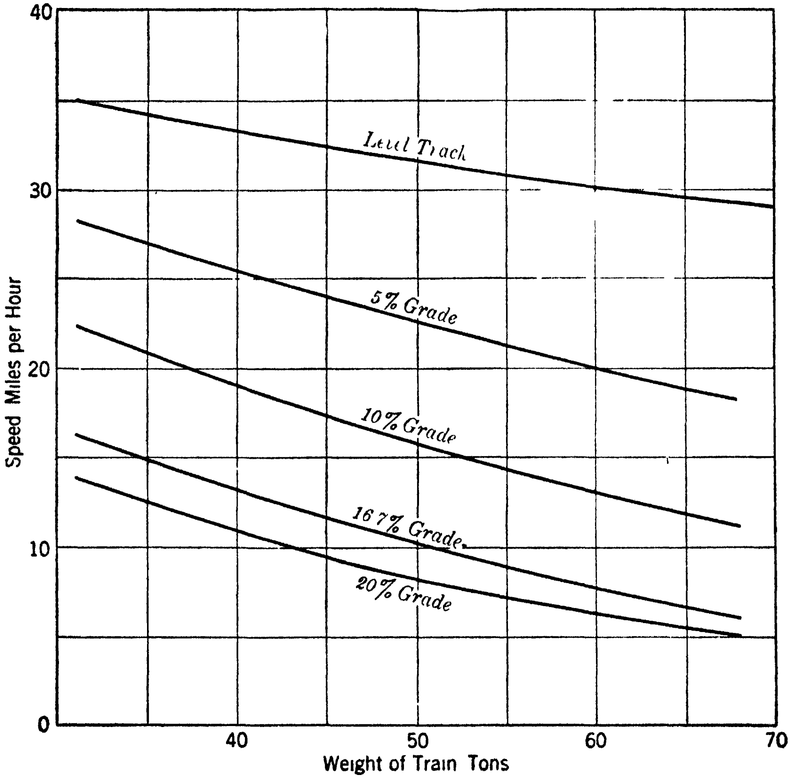

Fig. 325.—Efficiency Curves

starting the engine as well as running the car and charging the storage battery. Both the axle motors are standard railway type and drive the motor axles by means of single spur gears, which are enclosed in a tight gear case filled with grease. A defective motor can be quickly disconnected by means of a switch conveniently located in the cab of the motorman and the remaining motor used for driving the car.

The efficiency of the machinery is shown in figures 325, 326 and 327, showing the speed obtained on the level track as well as on different grades with different types of cars.

Type No. 1 on the level track—33 miles an hour
with 5% grade —24.2 " " "
with 10% grade —17.4 " " "
with 16.7% grade—10.4 " " "

With Type No. 2 with the same size engine but a narrow gauge appreciably greater speeds are obtained.

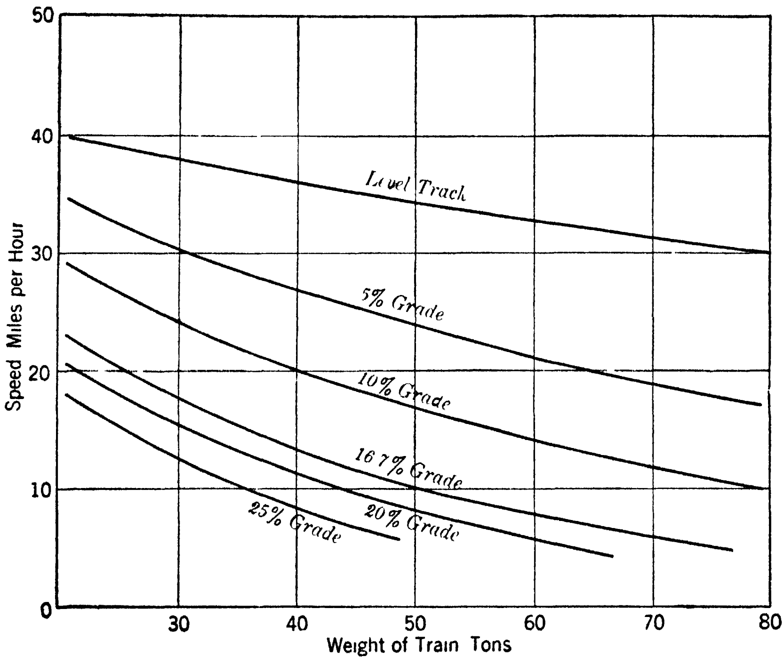

Fig. 326.—Efficiency Curves

Control. The cabs for the motorman are at each end of the car and are provided with all the necessary apparatus and instruments for the operation of the train such as the controller handle and maneuvering gear for the hand and air brakes, whistles and sand spreaders and an ammeter for the main generator and gauges for the air brake, injection air pressure and lubricating oil. A speedometer is also fitted showing the

speed of the train. The maneuvering controls have two drums, one for reversing and one for the speed of the car.

The reversing handle is removable in the zero position only and the speed changing drum through a special arrangement is thereby located so that no maneuvering can be made. The

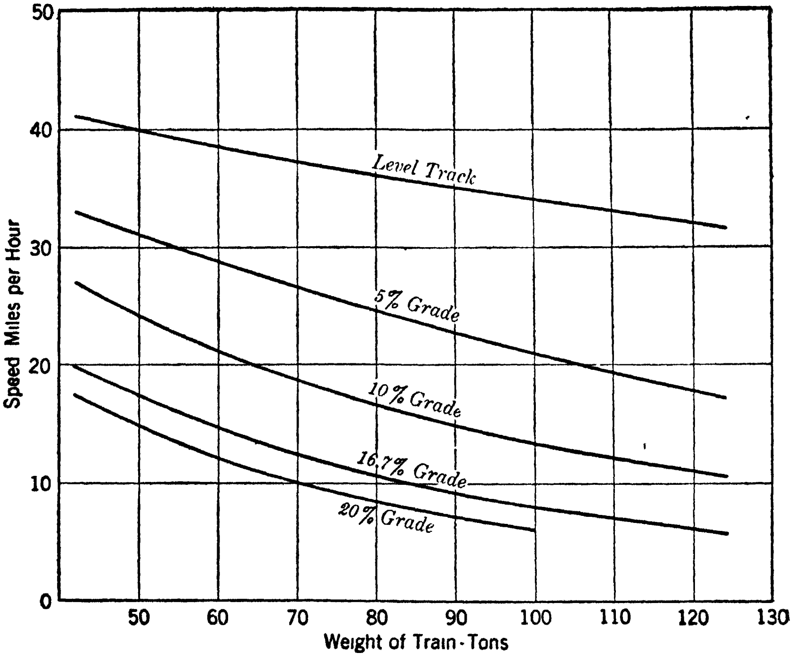

Fig. 327.—Efficiency Curves

motorman removes the maneuvering cap so that no maneuvering can be made by unauthorized people.

The Diesel engine is started by the maneuvering drum, which connects the axle motors and controls the speed. The control of the handle of this drum is provided with a safety push button or so-called dead man's grip, which must be held down while running. Immediately the push button is released the engine stops. This latter is provided so that should anything happen to the motorman and he releases the push button, the engine will stop and as no power is furnished to the

motor the train will consequently come to a standstill as soon as the inertia will allow. The train is operated as follows:

The crank for going forward or reverse is put into position for the desired movement of the train. The safety button is pushed down and the maneuvering handle put in position 1, at which the Diesel engine is started. By turning the crank to the next position the starting mechanism disengages and the axle motors coupled in start the train. Turning the handle still further increases the voltage and also the speed of the train. To stop the train the safety button is released and the maneuvering handle returned to starting position. Very little manipulation is necessary while the train is in motion on account of the characteristic of the electrical machinery to automatically maintain the load constant within a large range of speed of the train. This feature is especially desirable as it enables the motorman to give practically his whole attention to the track and the signals. The operation of the train, therefore, is practically similar to the running of an electric street car. While the Diesel unit is running and delivering current the revolutions remain constant and a great advantage with these cars is that when electric current is not needed, as for instance on going down a grade or when standing at the stations, the Diesel unit is stationary, but immediately same is needed for power, the unit is started electrically and is ready for operation in a few seconds. Through this feature a great saving is made, both in fuel and lubricating oil. A storage battery is usually hung under the car to furnish current at constant voltage for starting purposes, lighting and other auxiliaries. While the train is in motion, the storage battery is charged automatically but it can also be charged while the train is standing but with the engine running.

Maintenance and Upkeep. Some figures regarding the maintenance and upkeep of the general cost of operation are given below on this matter. Experience has shown that the maintenance of the cars is very simple and inexpensive, practically only the Diesel engine requiring any attention. The schedule, which has been followed out for some years with entire satisfaction, is to make examinations at stated inter-

vals, for instance, the fuel valves twice a month, compressor valves once a month, cylinder heads once a month, inlet and exhaust valves every two months, main pistons every three or four months, compressor pistons every three or four months, connecting rod ends every three or four months and the main bearings annually. As the valves are fitted into interchangeable cages, it is only a matter of a few minutes' work to change these. Therefore, it is only essential to put the car into the shops when it is necessary to examine the cylinder heads and the pistons. Usually a general overhaul is made after a distance of 37,000 miles has been run. This represents an average year's work in the Swedish service. It is, however, the almost unanimous opinion of the engineers in charge of these cars that the general inspection would be sufficient if it was made after every 60,000 miles instead of 37,000 miles. The results, however, obtained from the six different roads with six motor cars were surprisingly uniform and they are summed up in the following table.

It should be mentioned that one car has run for over four years, one for three and one-half years, one for two and one-half years, one for one year and two slightly over a half year.

Miles traveled since traffic started	410,000
Number of running days	4,380
Miles without trailer	246,000
" with "	164,000
" per day	93
Ton miles	15,650,000
Average weight of train, tons	38.3
Fuel consumption including charging storage battery, lbs. per ton miles	.031
Lubricating oil consumption, lbs. per mile	.042
Various materials, cents per mile	.0735
Train crew, cents per mile	3.44
Sweeping, cleaning, etc., cents per mile	.259
Maintenance and administration, cents per mile	2.08

The results obtained with the Swedish Diesel electric motor cars are very gratifying when compared with the results obtained with the other cars. The consumption of the

Diesel electric train varies from .0118 pound per ton mile with a train of 84 short tons average 26 miles per hour for a distance of 52 miles to a consumption of .0358 pound per ton mile with a 36 ton train averaging 27 miles per hour over a distance of 30 miles. In the first case one stop only was made; in the second there were two stops. In both cases the maximum grade was not over 17% but the profiles were different. The mean of the fuel consumption figures per ton mile for

Fig. 328.—Swedish Diesel-Electric Train

all the various stages was .0212 pound, but this figure of course must be accepted only for purposes of general indication and for the particular route followed with its varying levels and grades. The maximum speed shown on the chart was 31 miles per hour with six stops in a distance of 75 miles, the train weight being 36 tons and a maximum grade of 10%. The minimum speed recorded was 16 miles per hour over a distance of 3 miles without stops, the train being loaded to 66 tons and the maximum grade 20% over a distance of 2,200 miles. The fuel consumed by a Diesel electric train is only

6% of the weight of the total amount of coal burned by a steam train that runs regularly over the various sections of the road. In other words, the steam operation requires 17 tons of fuel where the Diesel electric system only requires one ton. The results of an actual test made of a Diesel electric train consisting of a motor car, Fig. 328, one 8-wheel car and one 4-wheel car as trailers, is as follows:

The train left Stockholm with 139 passengers. Some of these left and others boarded the train during the trip.

Weight of motor car	36.1 Tons
Weight of 8 wheel car	26.2 "
Weight of 4 wheel car	17.6 "
139 passengers @ 165 lbs.	11.5 "
Total weight	91.4 Tons

The results during the test run are shown below:

	Stockholm-Vasteras	Vasteras-Stockholm
Distance in miles	69	69
Number of station stops	8	8
Running time of train not including stops	3 Hr. 1 Min. 30 Sec.	2 Hr. 55 Min. 30 Sec.
Running time of the motor	1 Hr. 59 Min.	2 Hr. 11 Min.
Motor running in % of train running time	65.5%	74.5%
Average speed in miles per hour	22.9	23.73
Fuel consumption during trip in gallons	15.35	15.10
Spec. weight of fuel oil	.815	.815
Fuel oil consumption during trip, lbs.	104.5	102.2
Fuel oil consumption per mile, lbs.	1.515	1.48
Fuel oil consumption per ton mile if the average weight of train is 88 tons, lbs.	.01722	.01685

As mentioned earlier during water brake test of engine the fuel consumption was .428 lb. per brake horsepower of 50.6 lbs. per hour. On the above basis if the engine had been running with full load the fuel consumption of Stockholm-Vasteras would have been 101.2 lbs. and the Vasteras-Stockholm, 110 lbs.

The actual consumption was 104.5 lbs. and 102.3 lbs. respectively which agrees very well.

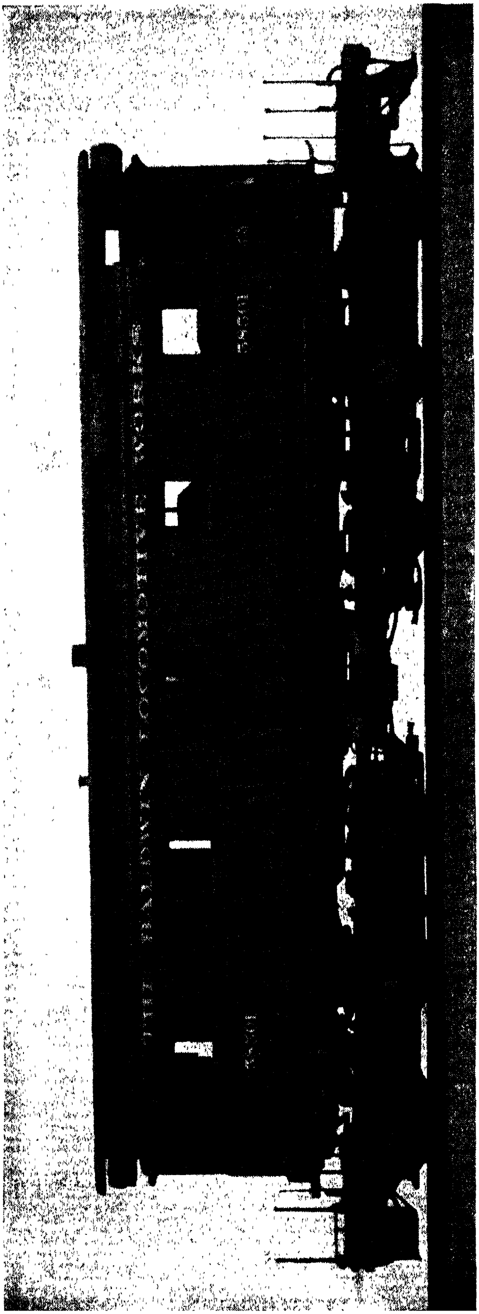

Fig. 329.—Baldwin-Knudsen Diesel Locomotive

From the foregoing it is evident that the Diesel electric railway car is a factor to be reckoned with, if only from the economy of performance that has been shown in the figures given. With the supply of fuel oil that there is in this country and with the indications of its remaining at a reasonably normal level, for some time in the future, oil engines which have already taken a very prominent part in our industrial development are destined soon to enter railway service, particularly in the manner described.

There is not much possibility of a rivalry between the electric and the Diesel electric systems. This seems to be very remote as after this question is fully analyzed it will easily be concluded that one system does not displace the other, rather they co-ordinate one with the other.

Due to persistent efforts to improve the thermal efficiency of the internal combustion engine, highly satisfactory results are obtained with the Diesel type of prime mover. The locomotive, however, still stands on a comparatively low level in this respect. The thermal efficiency of this unit under the most favorable circumstances is rarely as good as 8%, whereas the thermal efficiency of the Diesel engine easily averages 30%. This necessarily must play an important part from an economic point of view and is also easily borne out by comparison of the operating cost of steam and Diesel electric driven trains given below:

Comparative Figures

STEAM TRAIN COSTS: 64 PER CENT HIGHER
STEAM PASSENGER-MILE COSTS: 135 PER CENT GREATER

Comparison of Operating Costs—Swedish Basis (1914)
Annual Distance = 37,000 miles.

Mean speed = 17½ *miles per hour.*

STEAM TRAIN		DIESEL-ELECTRIC TRAIN	
Locomotive and tender...	50 tons	Car Type 1-43 seats......	32 tons
Trailer—58 seats	30 tons	Trailer—40 seats.........	11 tons
Baggage and mail car...	9 tons	Baggage and Mail—in coach	6 tons
Passengers and baggage..	6 tons	Passengers and baggage..	8 tons
Total weight 95	tons	Total weight	57 tons

1. LOCOMOTIVE SERVICE

	U. S. cents per mile
Engineer and fireman	4.16
Cleaners	0.77
Coal and water handling	0.35
Coal—24.8 lbs. @ $4.43 per short ton	5.49
Lub	0.21
Sundries	0.17
Locomotive sheds	0.55
Repairs	2.50
	14.20

2. CAR SERVICE

Cleaning, lighting & maintenance: 6 axles @ 0.308c. per train mile	1.85
Direct Operating Cost (cents)	16.05

3. INTEREST & DEPRECIATION

Locomotive:	
Capital stock $14,500	
Interest 5% & depreciation 5%	3.92
Trailers:	
Capital cost $8,300	
Interest 5% & depreciation 4%	2.00
Total Operating Cost (cents)	21.97

1. DIESEL-ELECTRIC SERVICE

	U. S. cents per mile
Motorman	2.51
Cleaning	0.20
Coal and water handling	nil
Fuel oil—1.5 lb. @ $30.80 per short ton	2.31
Lub	0.26
Sundries	0.05
Sheds	0.43
Maintenance	1.80
	7.56

2. CAR SERVICE

Cleaning, lighting & maintenance: 2 axles @ 0.334c. per train mile	0.67
Direct Operating Cost (cents)	8.23

3. INTEREST & DEPRECIATION

Motor coach:	
Capital cost $16,600	
Interest 5% & depreciation 5%	4.46
Trailer:	
Capital cost $2,400	
Interest 5% & depreciation 4%	0.58
Total Operating Cost (cents)	13.27

Baldwin-Knudsen Diesel Locomotive. The Baldwin Locomotive Co., in conjunction with the Knudsen Motor Corporation, have recently built a Diesel locomotive of 1,000 horsepower, which is illustrated in Fig. 329.

The engine of this locomotive is of unusual design, in that it is built on the lines of an inverted "V" with cylinders of 9¾ inches bore by 13½ inches stroke. The fundamental feature of this engine is its two-cycle operation, which is further characterized by the fact that the combustion spaces of each pair of cylinders forming the "V" are joined to form a single cavity. The dual crankshafts are geared together in such a way to make the pistons travel up and down approximately together, so that the compression and expansion parts of the cycle are practically the same as in an engine of the conven-

tional type. The crankshafts and gearing are shown in Fig. 330.

Port Scavenging. Ports located in the cylinders permit the introduction of scavenging air and the escape of the exhaust gases, the respective ports being located on opposite sides of the engine, with the result that the sweeping out of the spent gases is accomplished by a direct flow of air. The geared relation of the two crankshafts is of course one to one, but the meshing of the gear teeth is so chosen that the exhaust piston uncovers its ports somewhat earlier than the inlet piston, and has already closed them again while there is still a certain amount of opening left on the inlet side.

During the period when the ports on both sides are open the air has a direct sweep through, and from the time that the exhaust opens until closure is affected on the inlet side the full pressure of air used for scavenging is allowed to build up.

The scavenging air is furnished by a General Electric Co. turbo-blower driven at 3600 R.P.M. by the engines through gearing and is rated to deliver 4500 cubic feet of air at 2½ pounds' pressure. This blower consumes approximately 70 H.P., but this supercharging feature made possible by the use of the blower provides the cylinders with an extra quantity of fresh air, and in so doing adds to their power as though the bore and stroke and weight of the engine had been increased. A cross section of this engine is shown in Fig. 331 from which it will be seen that the piston pin is closed off from the upper end of the piston by means of a dome. In this piston it will be seen that a long narrow port cuts through the wall and communicates with the space between the dome and the underside of the piston crown.

During the scavenging period a part of the air current is diverted through these cored passages in the way of the cylinder ports in such a manner that the air present inside the hollow piston is replaced by a charge of considerably cooler air which thereupon absorbs heat until it is again forced out.

Fuel is injected by means of individual pumps at a pressure round about 7,000 lbs. per square inch, and the jet from the spray valve is so divided that each of the twin cylinders gets its proper share.

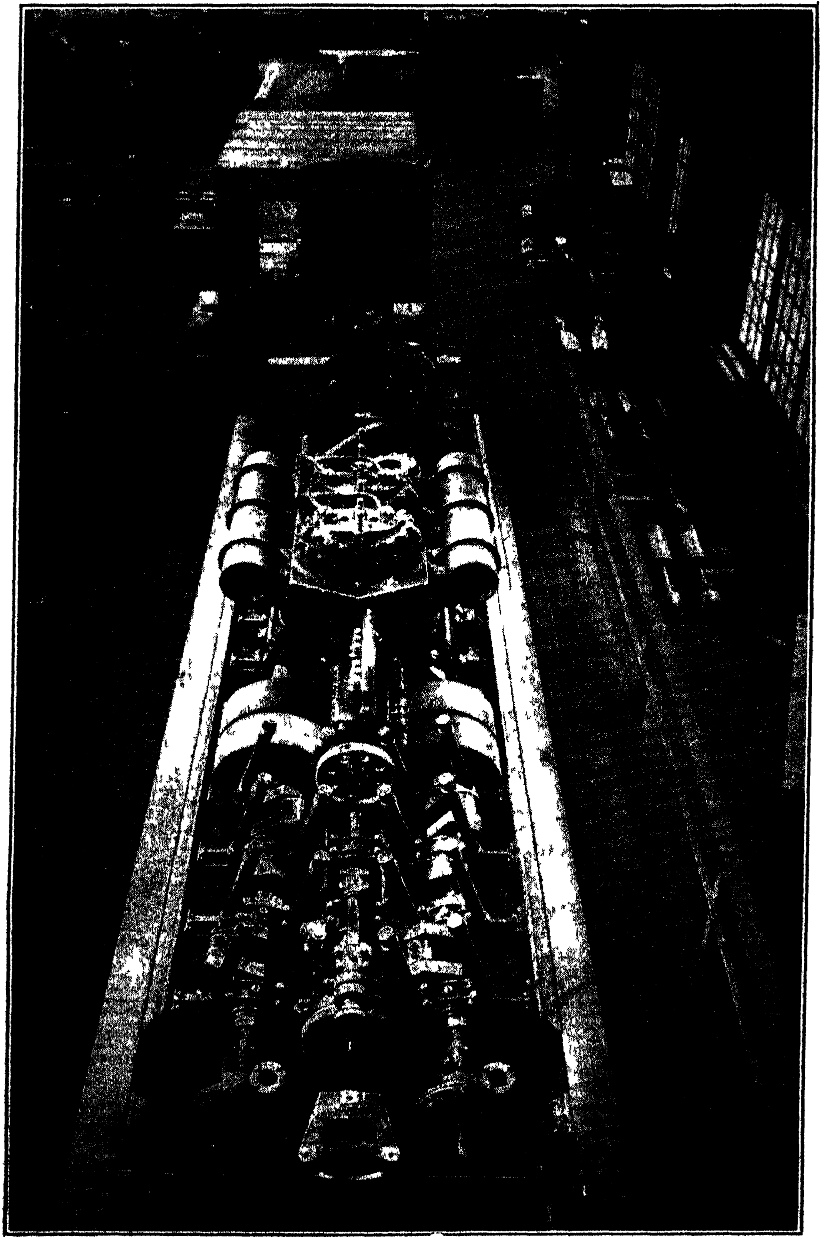

Fig. 330.—Dual Crankshafts and Gearing of Baldwin-Knudsen Locomotive

Twin centrifugal pumps driven from the engine by the same gearing system as is used for the blower, circulate the cooling water through an air cooled radiator resembling those of the automobile type and which is situated at one end of the locomotive as shown in Fig. 332.

Mounted upon each end of the twin engine shaft is a large

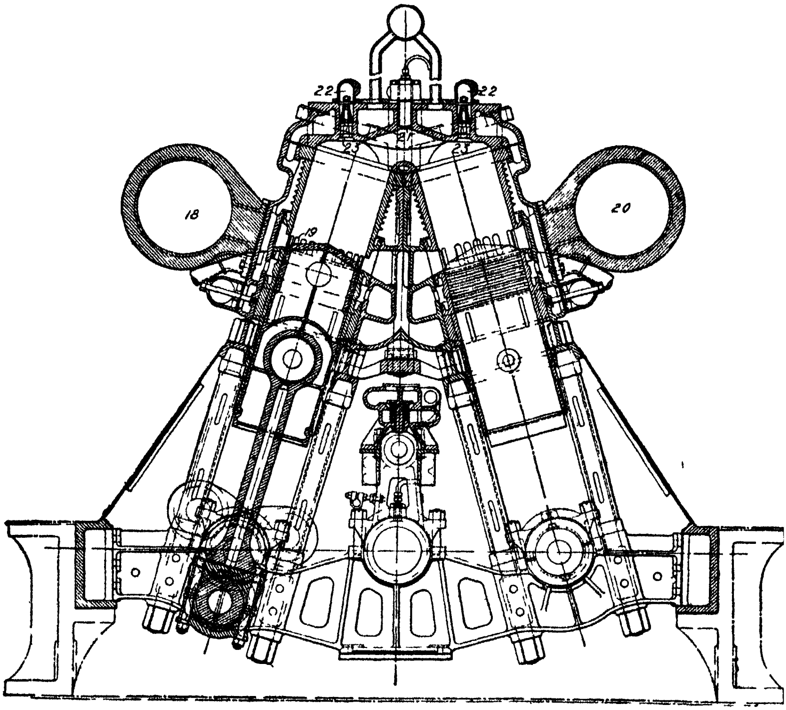

Fig. 331.—Cross-Section of 1,000 H.P. Baldwin-Knudsen Diesel Engine

double helical gear possessing enough inertia to act as a fly-wheel, while both the gears mesh with a common pinion keyed to the generator shaft. The crankshaft turns at 450 R.P.M. and the generator pinion at 1200 R.P.M. This speed allows a considerably smaller electrical machine to be used than would have been the case with a direct connected unit, and in view of the fact that locomotive designers demand a power plant of light weight this latter feature is of course advantageous

in this respect. The main generator fields are excited by a small separate exciter, also driven by gears. Simple controllers are located at the front and rear of the locomotive, making it

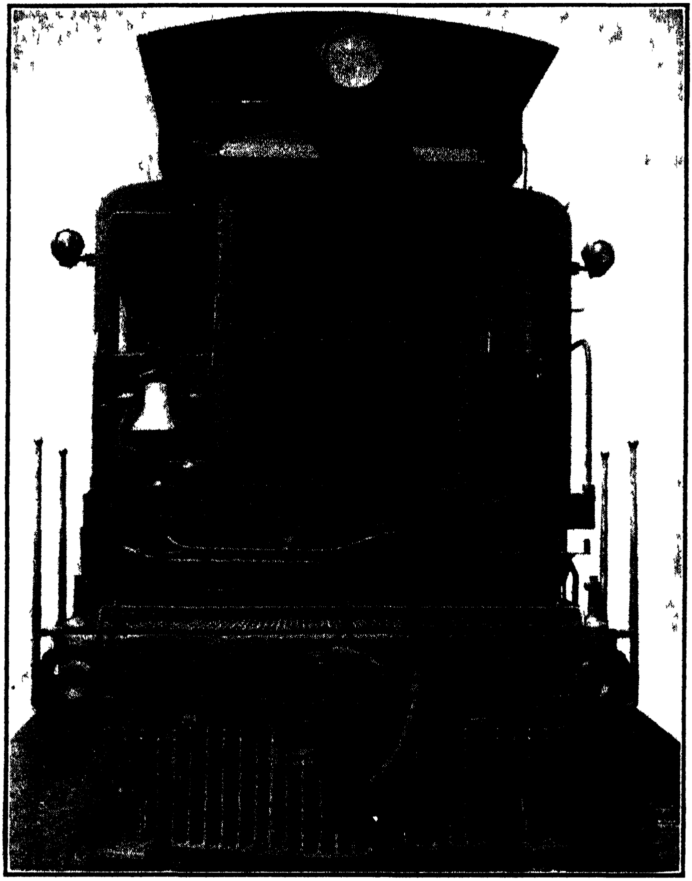

Fig. 332.—Air Cooled Radiator of Baldwin-Knudsen Locomotive

possible for the engineer to maneuver from either end with no more effort than it takes to operate a steam locomotive.

The system of control is of the well-known Ward-Leonard type, which makes it unnecessary to provide the heavy switches, rheostats and other electrical gear such as would

have to be fitted if the main power current flowing through the generators and traction motors had to be directly altered for the purposes of control.

According to the speed and power demands made on the locomotive the operator merely has to vary the relatively slight exciting current, while the reversal of the latter causes the locomotive to travel in the reverse direction. Linked with the controller handle is also the throttle control of the engine by means of which the power output is automatically varied according to the demands of the electrical transmission.

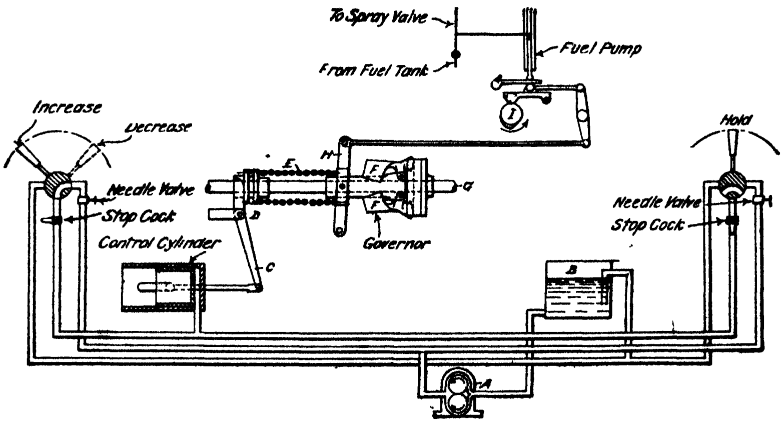

Fig. 333.—Diagrammatic Representation of Oil Supply Control of Baldwin-Knudsen Diesel Locomotive

The method of delivering the oil to the cylinders in the varying proportions depended upon the speed of the engine and the work it is called upon to perform is interesting and unique. The speed of the engine is controlled by a governor of the ordinary centrifugal ball type, the balls acting against a helical spring, the tension of which can be varied, the greater the tension that is put upon the spring the higher will be the velocity of the balls before a cutting off of the oil supply takes place. The arrangement is shown diagramatically in Fig. 333. In this there is an oil pump A deriving its supply from the tank B, and delivering it, under pressure, through the central pipe of the system to the needle valve at either

end of the car. The needle valve is adjusted so that the proper pressure will be maintained in or against the control valve.

In the diagramatic arrangement the control valve is shown in the holding position at the right hand end of the engine. That is to say, the entrance of the delivery pipe is blank and no oil can enter the valve.

At the left hand end of the engine the control valve is shown as connecting the delivery pipe with the central pipe which has a branch leading from it to the end of the control

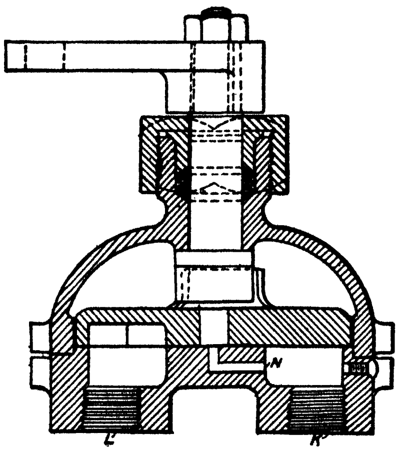

Fig. 334.—Oil Control Valve of the Governor

cylinder. It will be seen that the corresponding pipe leading from the control valve at the other end of the engine is blanked.

With pressure admitted to the right hand end of the control cylinder, the piston is moved to the left, carrying the lower end of lever C with it. This lever is pivoted at "D," and, as the lower end is moved to the left, its upper end compresses the governor spring "E." It is evident, then, that the greater this compression, the higher must be the rate of revolution of the shaft "G," in order that the governor balls "F" may move outwardly, and by a further compression of the spring so move the lever "H" so as to affect the fuel delivery apparatus at "I."

If the handle of the control valve is moved to the decrease position as indicated by the dotted lines the cavity of the valve will connect the pipe leading to the control cylinder with the one entering the valve at the left, permitting the oil to flow back through the lower pipe to the supply tank "B."

Control. The construction of the control valve is shown in vertical section and also with its valve face in Fig. 334, and also with its valve face in the three positions of "increase," "holding" and "off," in Fig. 335. To all intents and purposes this is merely a balanced disc valve operating as a three way cock. The oil enters under pressure at "K," flows through

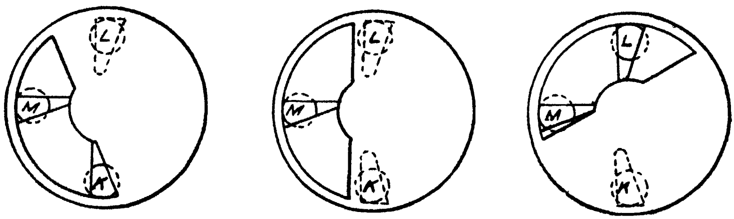

Fig. 335.—Seat of Oil Control Valve of Governor

the small hole end in the seat up through the center of the valve to the space in the casing above it. Here acting upon the whole area of the disc valve it holds it firmly to its seat. The three main positions of this valve are shown clearly in Fig. 335. In the increase position the oil enters at "K," flows through the cavity of the valve and out at "L," and thence to the control cylinder. In the hold position both the supply and exhaust openings are blank, and the piston of the control cylinder is held in place. With the valve in the off position, the oil flows back through "M" to the exhaust opening "L," and the tension on the governor spring is relaxed, so that it can resume its normal position and the governor affect reduction of the oil fed to the cylinders at the minimum speed of rotation.

Fuel Pump. The fuel pump in this engine is also worth consideration, and a section is shown in Fig. 336. Instead of a direct connection between the lever at the right of "I" in Fig. N and the slide No. 1 in Fig., the attachment is made at No. 2

at the lower end of lever No. 3 which is fastened to the same shaft as the mitre gear No. 4, which meshes with a similar gear keyed to shaft No. 5, and as it turns the lever No. 6 is moved to and fro and with it the slide No. 1.

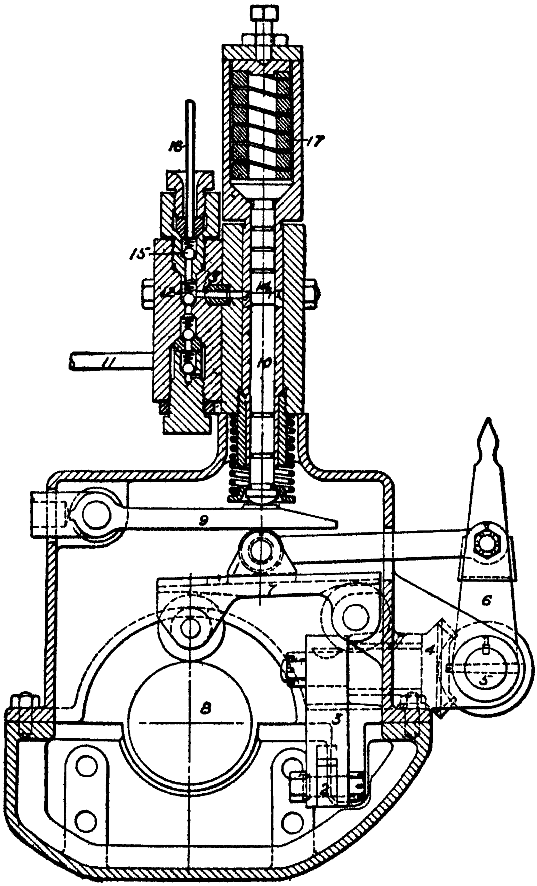

Fig. 336.—Fuel Pump of Baldwin-Knudsen Diesel Locomotive Engine

The cam arm No. 7 is raised and lowered at the proper moment by the rotating cam No. 8. As the cam arm is raised it lifts the arm No. 9 and with it the plunger No. 10 of the pump. The latter is held against the arm No. 9 by a helical spring as shown.

With the governor weights "in," the slide is directly beneath the centre of the pump plunger, and the latter is given its maximum stroke. As the speed of the engine increases and the governor balls fly outward, the slide is drawn to the right, thus decreasing the stroke of the pump plunger and delivering less oil to the cylinders.

The oil supply to the pump is supplied under pressure and controlled by ball valves as is shown in the section.

Two long steel channels form the backbone of the engine and provide a direct mounting for the engine and generator.

This engine is, of course, of relatively light weight and high speed, and appears to have gone a long way towards meeting the requirements of locomotive technicians calling for minimum weight per horse power and space occupied. The locomotive is carried on two six wheel trucks, each having a wheel base of 12′8″ with wheels of 40″ in diameter. The total wheel base is 38′4″ and the overall distance of the couplers 52″ 1¾″. The total weight is 275,000 lbs., of which 180,000 lbs. or a little less than two-thirds is upon the driving wheels. The starting tractive force is 52,200, which gives a ratio to the weight on the driving wheels of 1 to 3.44.

The main frame of the engine does not extend the whole length of the cab, but has a total length of 28′ or 2′4″ more than the distance between the truck centres which is 25′8″. The side sections of this frame are of cast steel 20″ deep with stiffening ribs at intervals. This frame serves not only as a base to carry the cab framing but also as a foundation for the engine frame.

The truck frames are of the regular locomotive type of cast steel with the usual pedestals and binders. The springs are placed over each oil box, the two inner ones, which are for the driving wheels, are equalized together while the outer spring is attached to the frame at each end. A novelty has been introduced in these springs by making each of them one-half of the width required to carry the load and placing them in pairs on each side of the frame. By this means an essential and desirable lowering of the cab framing was accomplished.

Ingersoll-Rand Oil-Electric Locomotive. A recent locomotive built by the combined efforts of the Ingersoll-Rand Co., the General Electric Co. and The American Locomotive Co., is shown in Fig. 337 and a sectional view showing the location of the equipment is shown in Fig. 338.

The Ingersoll-Rand oil engine used in this locomotive, Fig. 339 is of the vertical, six cylinder, four cycle, single act-

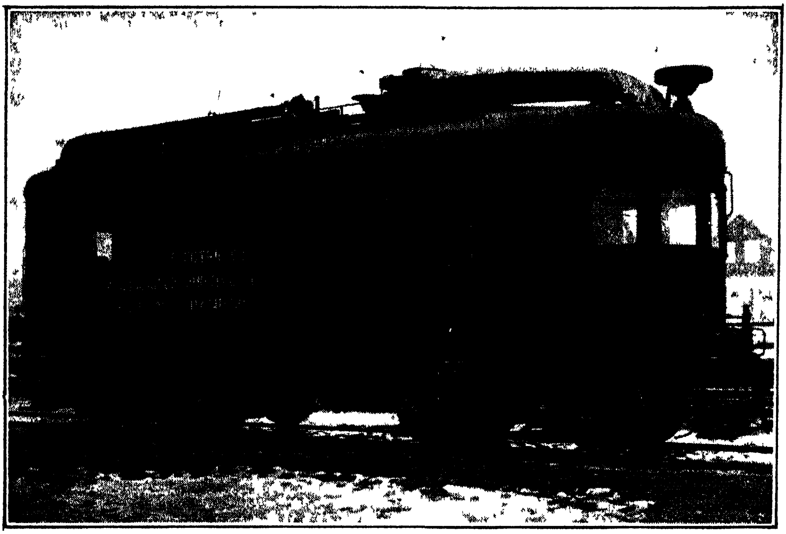

Fig. 337.—American Locomotive-Ingersoll-Rand Oil-Electric Locomotive

ing, solid injection, variable speed type. Fuel oil injection is accomplished by means of two opposed spray nozzles in each combustion chamber to which oil is delivered under pressure by an injection pump driven from the main shaft. No compressed air is used for fuel, injection and ignition is produced by the heat of compression only.

One fuel injection pump serves all cylinders. The fuel oil distribution is obtained by a distributor timed to admit oil to the spray nozzles of each cylinder in their proper firing order.

The lubricating system is entirely enclosed and is of the forced feed type. Lubricating oil is pumped to the main mov-

ing parts of the engine by a gear driven pump in the crank case. Oil in contact with the cylinder walls is passed through a filter and returned to the crank case oil reservoir.

The engine develops 300 B.H.P. at 600 R.P.M. and has a bore of 10 inches and stroke of 12 inches. A closed water cooling system is used on the engine. The water is circulated by a centrifugal pump driven from the crankshaft. The temperature of the water in the engine jackets is regulated by a thermostatic valve, which controls the circulation of the cooling water from the engine to fin tube radiators of 1200 square foot cooling surface located on the locomotive roof.

The engine is started by means of compressed air at about 200 pound pressure which is admitted to each cylinder in succession through mechanically operated starting valves. While in operation the engine drives a small air compressor which maintains pressure continuously in the starting reservoirs.

The generator is a 200 K.W., 600 volt, direct current, compound wound, commutating pole unit, separately excited. The generator, together with its exciter, is specifically designed for this service and is direct connected to the engine. The combined characteristics of generator and exciter are such as to produce a machine of practically constant output. The voltage of the generator is regulated by the current demand of the traction motors, so that, making due allowance for the generator losses, the product of this current and voltage is equal to the engine power. The K.W. output of the generator varies with the output of the engine, and at any position of the throttle it is constant throughout the whole working range of the power plant.

A 6 K.W., 60 volt exciter is mounted on the same shaft with the main generator and serves to excite the field windings of the main generator. A 32 volt storage battery is charged by this exciter through one of the field windings in series. The exciter and storage battery circuit is controlled automatically by a switch on the main throttle of the locomotive.

With this generator the control of the locomotive becomes extremely simple. There are two control handles. One is a throttle lever which controls the output of the engine and the

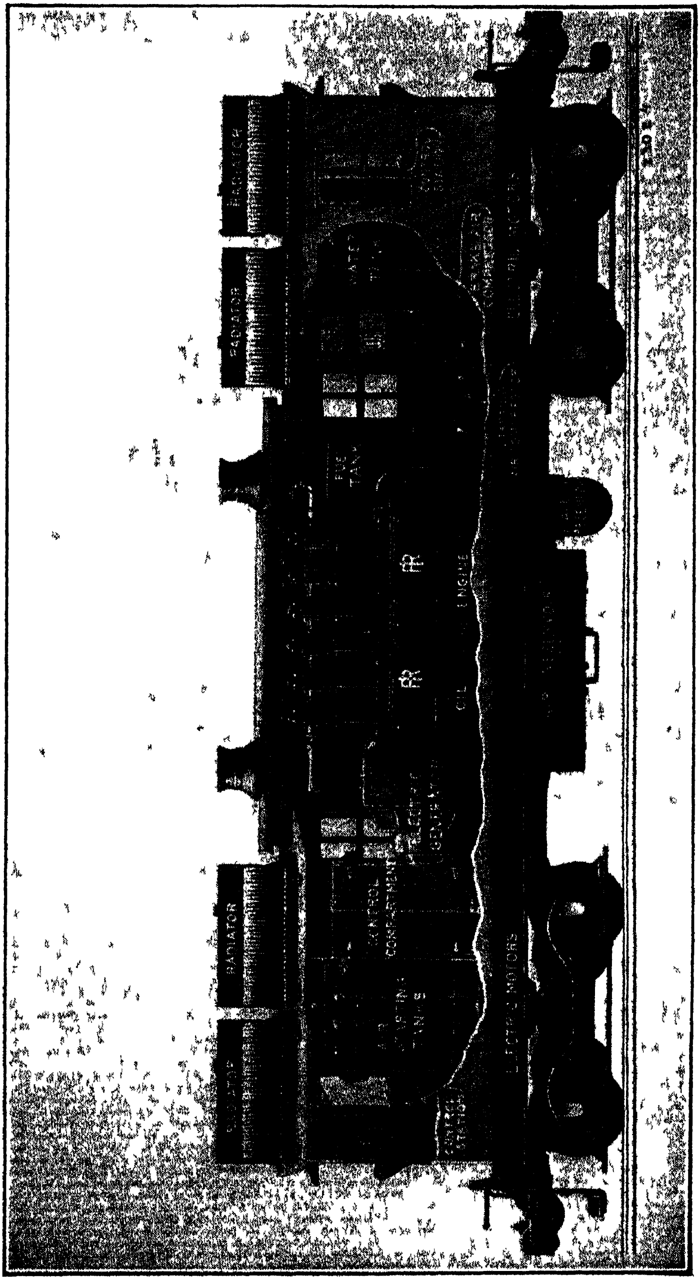

Fig. 338.—Sectional View American Locomotive-Ingersoll-Rand Locomotive

other is a master controller, or electric switch handle, which connects the traction motors in series or parallel for forward or backward motion. No rheostats are used in the power circuit which reduces to a minimum the loss of power during acceleration.

In operation, the electric control handle is set for forward or backward motion, with the motors in series for speed below 5 miles per hour or in parallel for speeds above 5 miles per hour. The position of the throttle lever determines the power delivered by the engine, and the generator and motors transmit this power to the driving wheels, automatically adjusting the proportion of tractive effort and speed to the load on the locomotive and automatically changing these proportions to suit the varying requirements of acceleration or grade.

The locomotive is equipped with four motors mounted on the trucks and geared to the driving axles. These motors are of the series wound, totally enclosed, commutating pole, split frame type; the axle, brackets and suspension lugs, being on the lower frame, render the motor readily accessible for inspection and repairs. A large hand hole, fitted with dust proof cover, is provided for on the commutator end through which the brushes and commutator may be inspected. The armature is carried in separate heads clamped between the motor frames and is provided with self-aligning frictionless bearings.

In the final tests it was proved that the oil-electric drive costs only one-third as much to operate as it does to run a steam switching locomotive for the same service.

In the course of six days' operating on the Central Railroad of New Jersey this locomotive handled and distributed 431 cars, moving them on and off 26 car floats.

The actual work was done in 60 hours and 50 minutes during which the locomotive consumed fuel. A steam locomotive, in the same service, handled and distributed 430 cars, also on and off 26 car floats, doing the work in 75 fuel-consuming hours. Incidentally, the steam locomotive stood by, ready for work with its fires banked for 69 additional hours, burning coal all that time. During this time the total fuel and lubri-

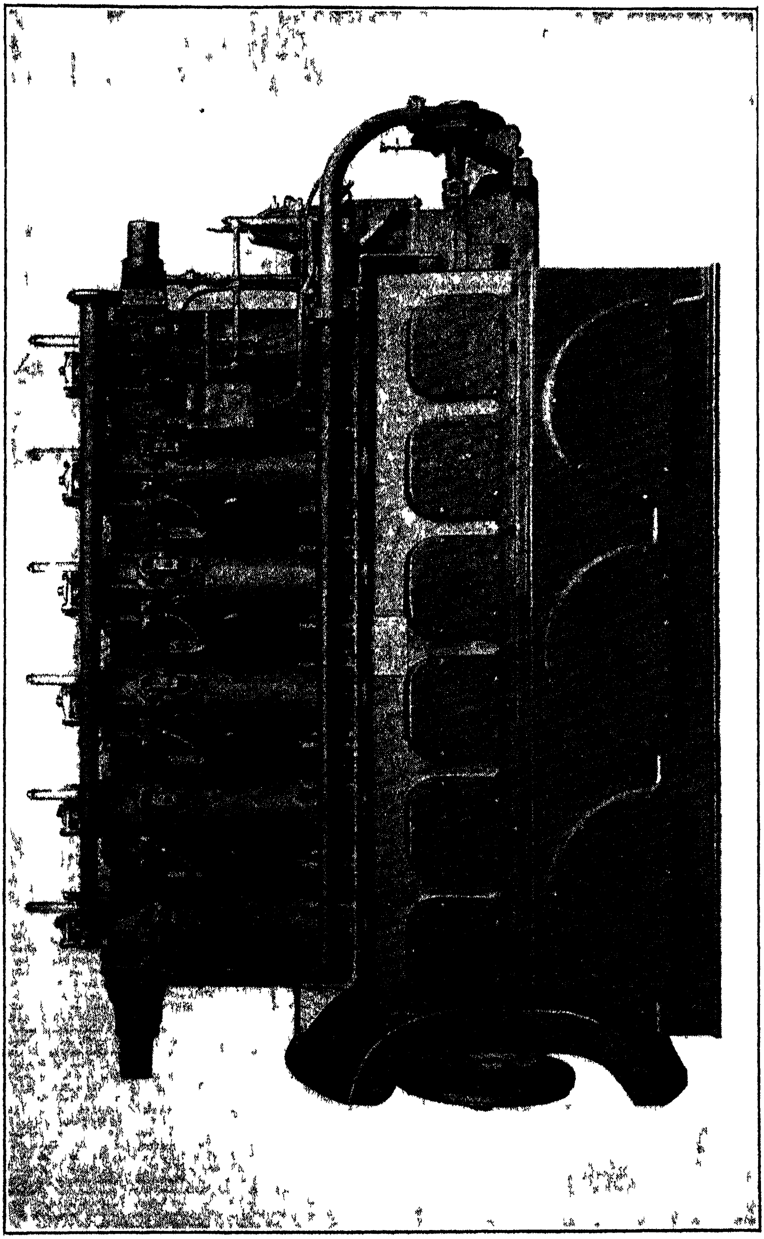

Fig. 339.—Ingersoll-Rand Locomotive Engine

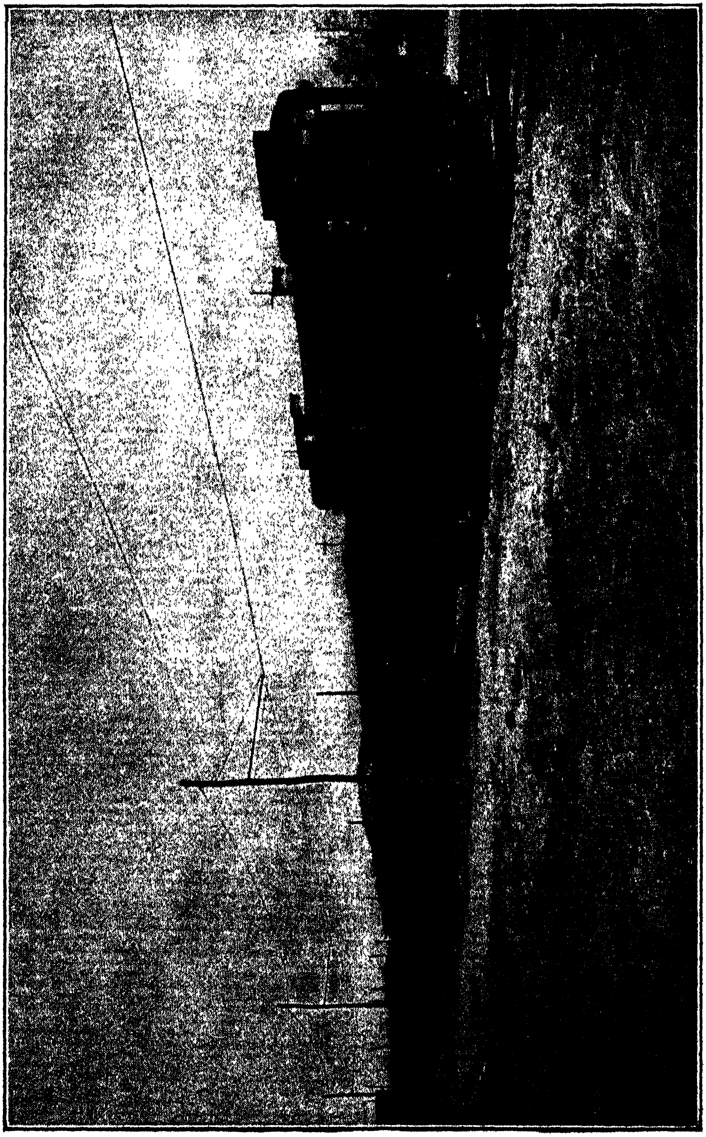

Fig. 340.—American Locomotive-Ingersoll-Rand Locomotive Under Test at a Long Island Freight Yard

cating oil cost of the steam locomotive was $73.35, while that of the oil-electric was only $11.90.

During a period of operation on the New York Central Railroad, this locomotive hauled 400,000 ton-miles and ran a total of 1,531 miles, giving an engine mileage of about 1.84 miles per hour for the time that the engine was in service, or about 2.64 miles per hour for the 579.33 hours that the oil engine was in operation.

On two occasions comparative tests were made between this locomotive and a Shay steam locomotive, both tests being made at the 33rd street New York Yards of the New York Central Railroad. The first test lasted one week and the second test two weeks. The results of each test were as follows:—

FIRST TEST

	Oil-Electric	Steam
Idle hours	9.58	14.66
Fuel consumed	153.5 gals.	7.5 tons coal
Lubricating oil used	14.5 gals.	1.28 gals.
Total fuel and water cost	$14.00	$86.00
Cost per hour of engine operation	$.366	$3.69
Cost per hour locomotive service	$.293	$1.695
Cost per 1000 miles	$.543	$3.82
Total ton-miles	25,905	22,510

SECOND TEST

	Oil-Electric	Steam
Hours of locomotive service	280	288
Fuel consumed	673.7 gals.	45.56 tons coal
Lubricating oil consumed	80. gals.	3.31 gals.
Total fuel and water cost	$73.00	$462.00
Cost per hour locomotive service	$.263	$1.695
Miles run	466.	445.6
Cost per mile	$.158	$1.04

During the past year that this locomotive has been in operation the data obtained compares very favorably with the results of the foregoing tests as regards the economy of working this type of engine.

The tractive effort of the oil-electric locomotive is different from the steam locomotive, in that the drop as the speed increases is more rapid. The accompanying tractive effort curve, Fig. 341, has been worked out for the power plant of this locomotive and shows the drop very effectively.

With the assumption that the maximum tractive effort available is 30 per cent of the weight of the locomotive it is

found that this is available only at speeds of .8 mile per hour or less. As the speed increases it falls off very rapidly and at a speed of 2 miles per hour it has fallen from the 30 per cent or 36,000 pounds to 21.7 per cent or 26,000 pounds. At 4 miles

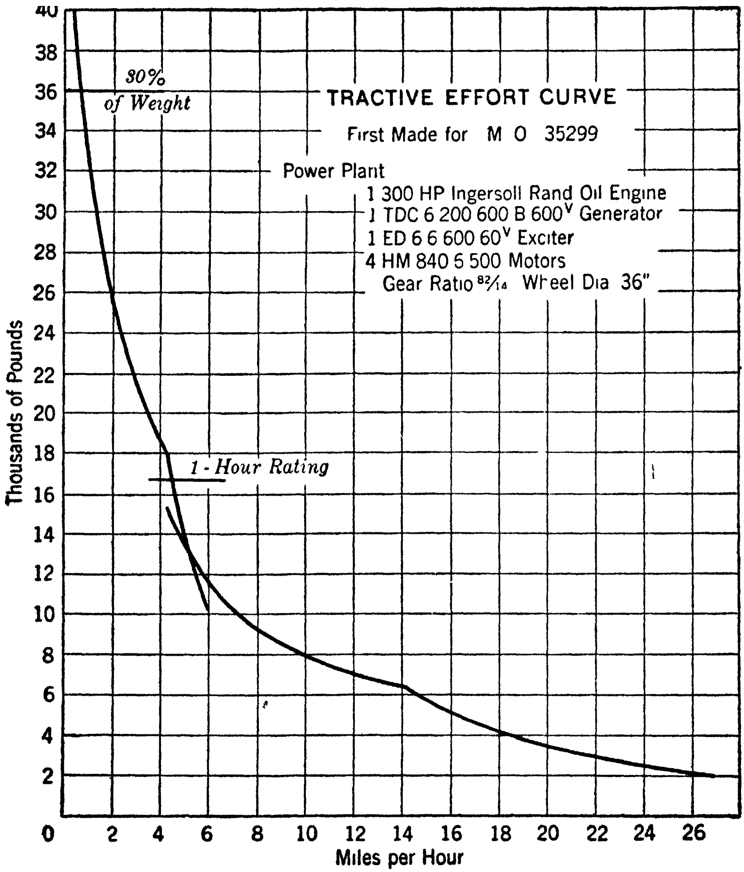

Fig. 341.—Tractive Effort Curve with 60-Ton Locomotive

per hour it has fallen to 18,000 pounds. This is on a one hour's rating. The fall from this point on is less rapid and at ten miles per hour it is 8,000 pounds and as the speed is still further increased it falls to 2,000 pounds at 27 miles per hour.

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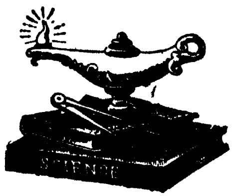

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